COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

May 1958

Pacific Gas and Electric Company's 660,000 kw Pittsburg Power Plant, to be enlarged by two 325,000 kw turbine-generator units for service in 1960–1961 respectively.

Packed Glands for High Pressures

APDA Reactor Vessel

Automatic Control for Precipitators

American Power Conference—II

The huge riverside crane at Combustion's Chattanooga Division, capable of lifting loads in excess of 250 tons, is about to ease the reactor vessel into a 195-ft. barge which will transport it via the Tennessee and Ohio Rivers to Cincinnati. From there it will travel by rail to the Enrico Fermi Atomic Power Plant at Lagoona Beach, Michigan — 30 miles below Detroit.

AMERICA'S LARGEST NUCLEAR COMPONENT

The nuclear reactor vessel shown here is the most remarkable stainless steel pressure vessel ever built. This 36-foot, 92-ton giant is the container for the nuclear furnace that will power the world's first full-scale, fast-breeder reactor power plant — the Enrico Fermi Atomic Power Plant. Designed by Atomic Power Development Associates, Inc., the nuclear section of this plant will be owned by Power Reactor Development Co., and the turbine-generator section by Detroit Edison Co.

Because of its vast size and complexity, the reactor vessel presented unique problems of design and fabrication. In reality, it comprises four separate cylindrical vessels welded together to form a single unit assembly, plus a large amount of internal shielding the fabrication of which involved the use of 65,000 square feet of stainless steel — all constructed and assembled with a precision never before attempted in work of this size and character. Additional parts of the vessel, scheduled for later shipment, will bring its total weight to about 200 tons.

No more than two or three plants in the world presently have the skills and facilities even to undertake the production of work of this kind. Combustion not only has the men and machines, but also the experience to produce such reactor vessels. Moreover, it is equipped and qualified to design and manufacture all other major components of complete nuclear power systems.

The Enrico Fermi Plant exemplifies the continuing, urgent drive by electric utility companies to generate electricity at the lowest possible cost, utilizing all developments modern science and technology can produce. Combustion is proud to have shared in this major step of the power industry to prepare itself for the age of the atom.

COMBUSTION ENGINEERING

Combustion Engineering Building • 200 Madison Avenue, New York 16, N. Y.

C-158

ALL TYPES OF STEAM GENERATING, FUEL BURNING AND RELATED EQUIPMENT; NUCLEAR REACTORS; PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 29 **No. 11**

May 1958

Feature Articles

Packed Glands for High Pressure: An Analysis of Fundamentals ..... *by Professor J. L. Thomson* 38

APDA Fast Breeder Reactor Vessel 53

Static Automatic Control For Electrical Precipitators ..... *by Larry L. Little* 55

American Power Conference in Review—II 59

Abstracts From The Technical Press, Abroad and Domestic 65

Editorials

Pioneering and Power Plants 37

Departments

Advertising Index 74, 75

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BPA

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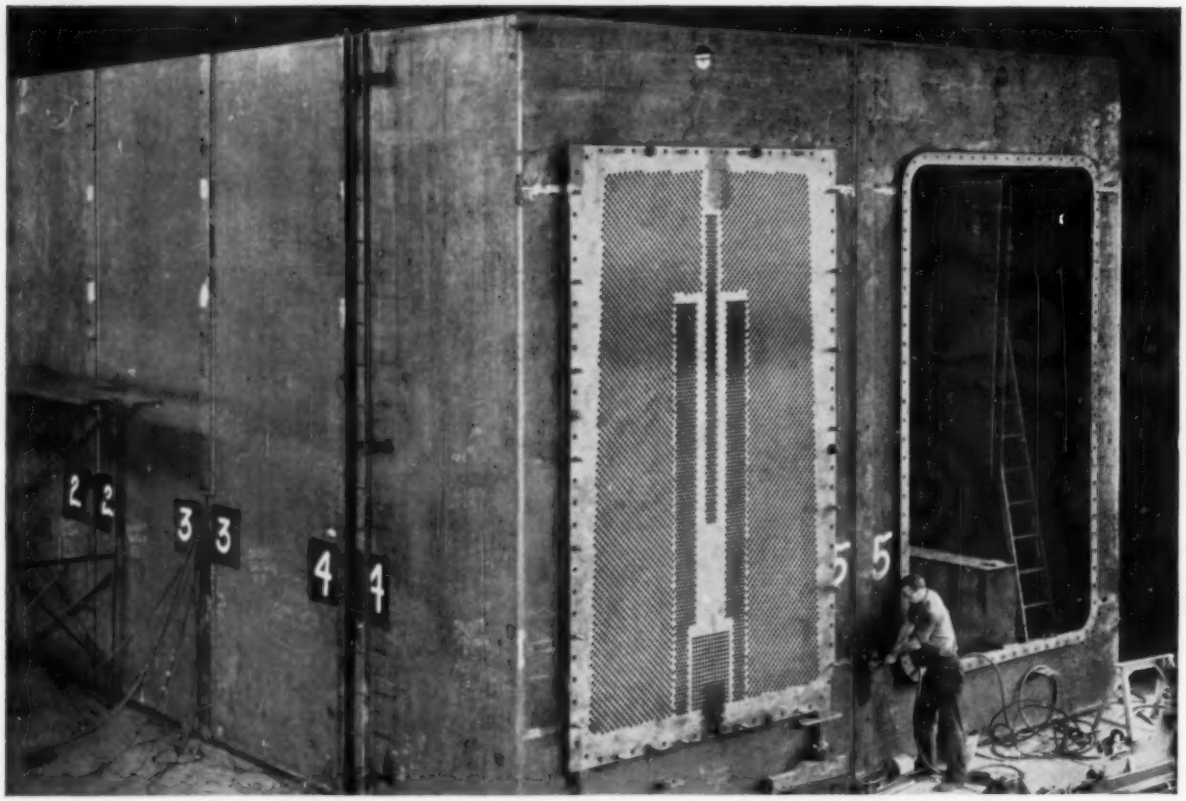
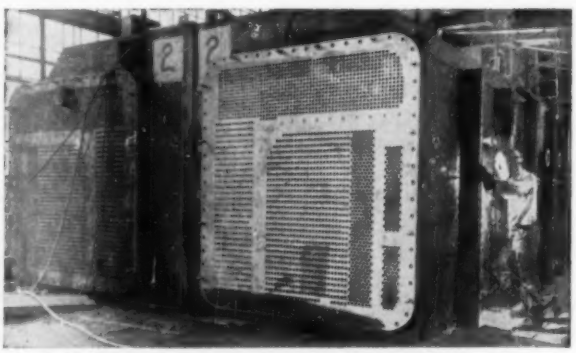


HERE'S ONE SHELL of this twin-shell, triple-lane unit just before shipment to Consolidated Edison's Astoria Station. It's designed to condense 1,600,000 lbs. steam/hr. at 1.87" Hg., with 244,000 gpm circulating water, and has 27,450 aluminum-brass tubes. Unit serves 335,000 kw turbine.

HUGE, SINGLE-PASS, 187,000 sq. ft. CONDENSER

... designed and built by C. H. Wheeler is now being installed at world's largest metropolitan utility

VICE-PRESIDENTS LEE YETTER and Roy Droescher, and Chief Engineer Paul Hamm are responsible for the design and construction of all Wheeler condensers. They work with engineers employed by C. H. Wheeler's customers, with turbine manufacturers' engineers and consulting engineers in BTU chasing.



TYPICAL REVERSE FLOW CONDENSER is this 35,000 sq. ft. unit for a Southern electric utility. Patented Reverse Flow feature permits flushing debris from tubes with only slight (and momentary) vacuum loss. Note low height to save head room, rectangular cross section to further utilize space for this Wheeler client.

C. H. Wheeler Mfg. Co. | 19TH & LEHIGH AVENUE PHILADELPHIA 32, PA.

Steam Condensers • Vacuum Equipment • Centrifugal, Axial & Mixed Flow Pumps • Marine Auxiliary Machinery • Nuclear Products

WELBOND®
is the valve
for high pressures
for high temperatures

Welbond Valves are winning acceptance among modern steam plants for all high pressure, high temperature valve service.

Superheater vents and drains, water wall drains, water column emergency shut-off, strainer blow-off, economizer drains—are just a few places where Yarway Welbond Valves are proving their mettle.

These special Welbond features insure improved valve performance:

- **Stem** of 321 stainless steel, used with special **packing** to prevent stem corrosion
- Easy and quick **accessibility.** All working parts removable through yoke. Jack action of stem forces out old packing
- Non-distorting, thermally-compensated **seat**
- Ventilated, easy grip **handwheel**
- **Streamlined flow** through body

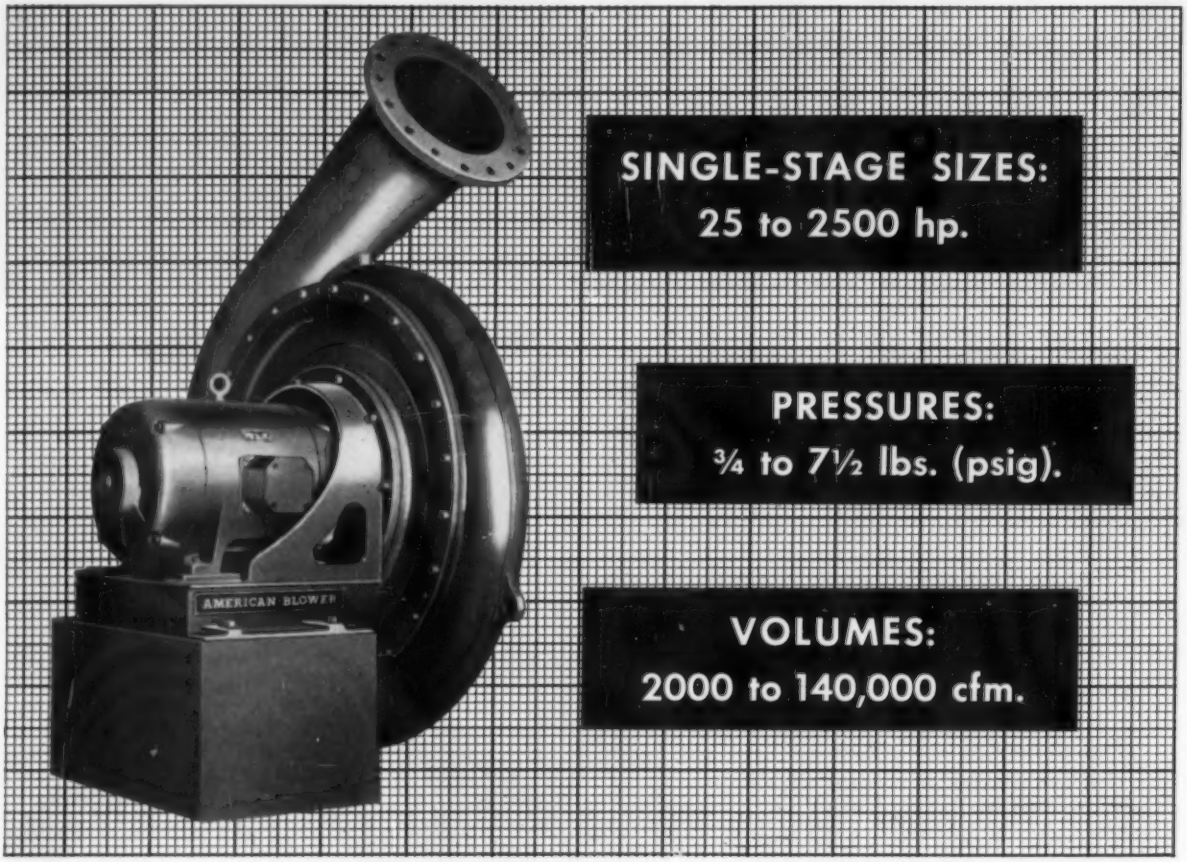
For full information, ask your Yarway man or write for Bulletin B-452.

YARNALL-WARING COMPANY
100 Mermaid Avenue, Philadelphia 18, Pa.
BRANCH OFFICES IN PRINCIPAL CITIES

Yarway Welbond Valves are available in nine sizes, ¼" to 2½", angle and straightway designs. Pressures to 2500 psi, temperatures to 1050° F.

YARWAY WELBOND ... *a good way to specify high pressure/high temperature valves*

American Blower Centrifugal Compressors



Rate centrifugal compressors for dependable performance, ability to meet specific pressure-volume requirements, minimum space and maintenance requirements. You'll find that American Blower single-stage units score high on all counts.

Adaptable to all drives, American Blower Centrifugal Compressors feature volute casings for increased efficiency. Compressor discharge can be located in increments of 15° in either direction of rotation, permitting great flexibility in application.

Compactness, high-speed operation, statically and dynamically balanced rotating elements minimize foundation requirements. Aerodynamic design of impeller blades contributes to efficiency, assures longer service life.

If you have an application calling for centrifugal compressors, talk to an American Blower sales engineer about your specific requirements. And check with him about American Blower Mechanical Draft Fans, Fly Ash Precipitators, *Gyrol*® Fluid Drives for boiler-feed-pump and fan control. His practical knowledge of air-handling and power-transmission problems may prove most helpful to you.

Call your nearby American Blower branch office, today. Or write: American-Standard,* American Blower Division, Detroit 32, Michigan. In Canada: Canadian Sirocco products, Windsor, Ontario.

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AMERICAN-Standard
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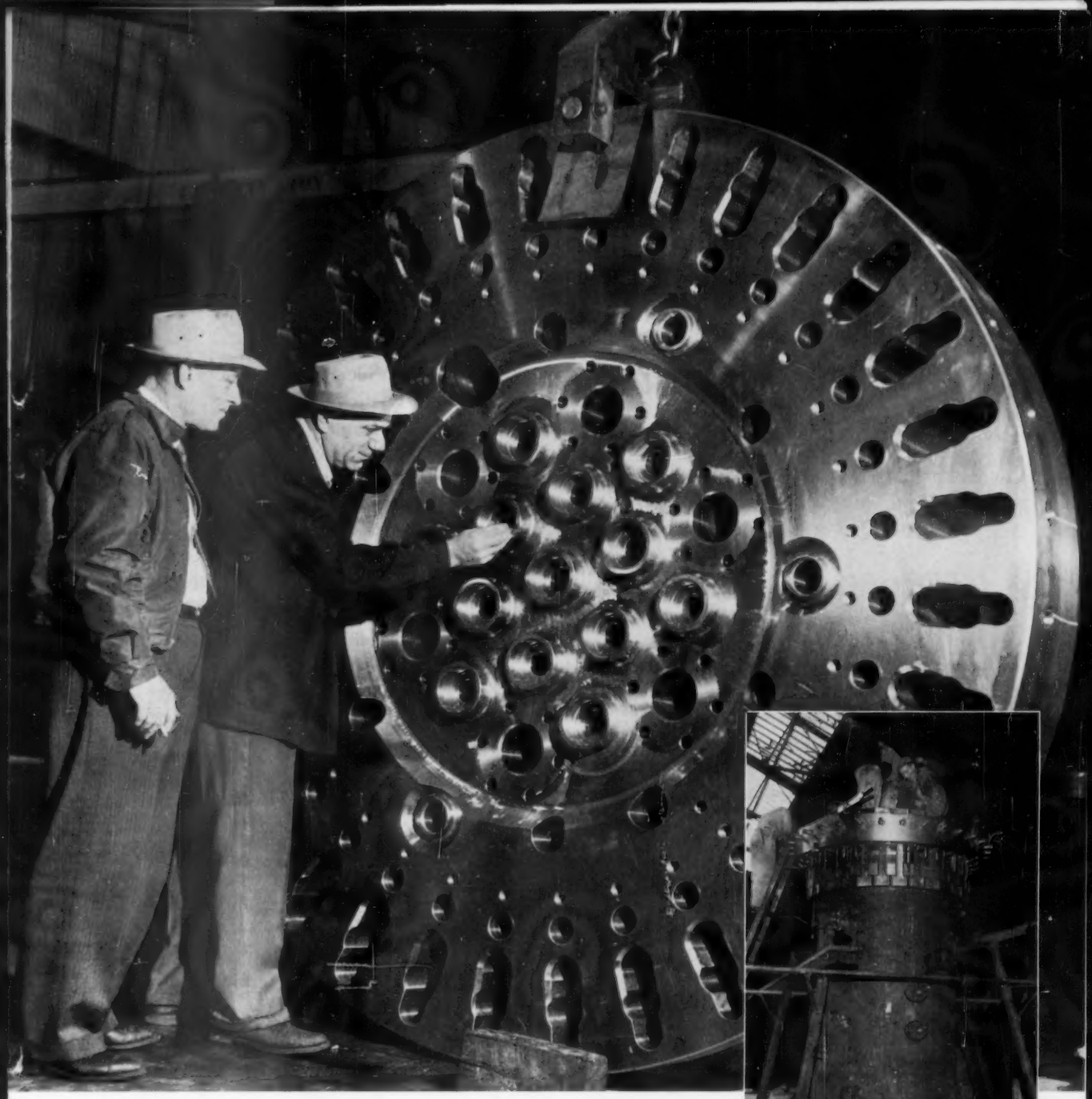

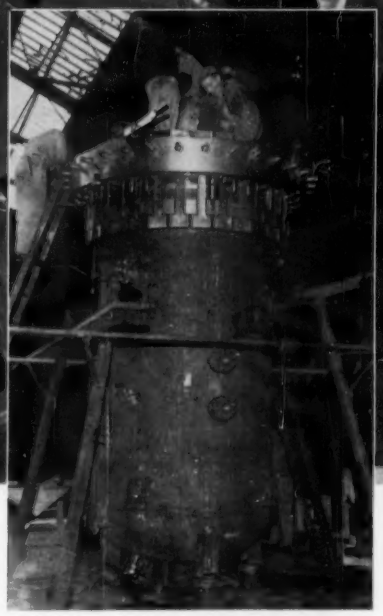

This pressure vessel had to be designed with a closure that would open in 30 minutes, and withstand 1500 psi, 550F. Made of Type 304 stainless, the top head has 34 connections, of which 19 required exceptionally close tolerances for control rods.

FOR NUCLEAR NEEDS of Knolls Atomic Power Laboratory, M. W. Kellogg was given the exacting assignment of designing and producing a 32½-ton "Proof Test Reactor Pressure Vessel". This was an unusually complex task, because of the quick-opening closure specifications, requiring a head with novel design and extremely close tolerance machining.

FOR NUCLEAR NEEDS of others, Kellogg is supplying the primary coolant stainless piping for two nuclear plants. In addition, Kellogg is equipped to engineer and manufacture heat exchanger equipment for nuclear energy power plants. If these specialized skills suggest a solution to *your* nuclear problems, call Kellogg's Fabricated Products Sales Division.

THE M. W. KELLOGG COMPANY, 711 THIRD AVENUE, NEW YORK 17.

A SUBSIDIARY OF PULLMAN INCORPORATED

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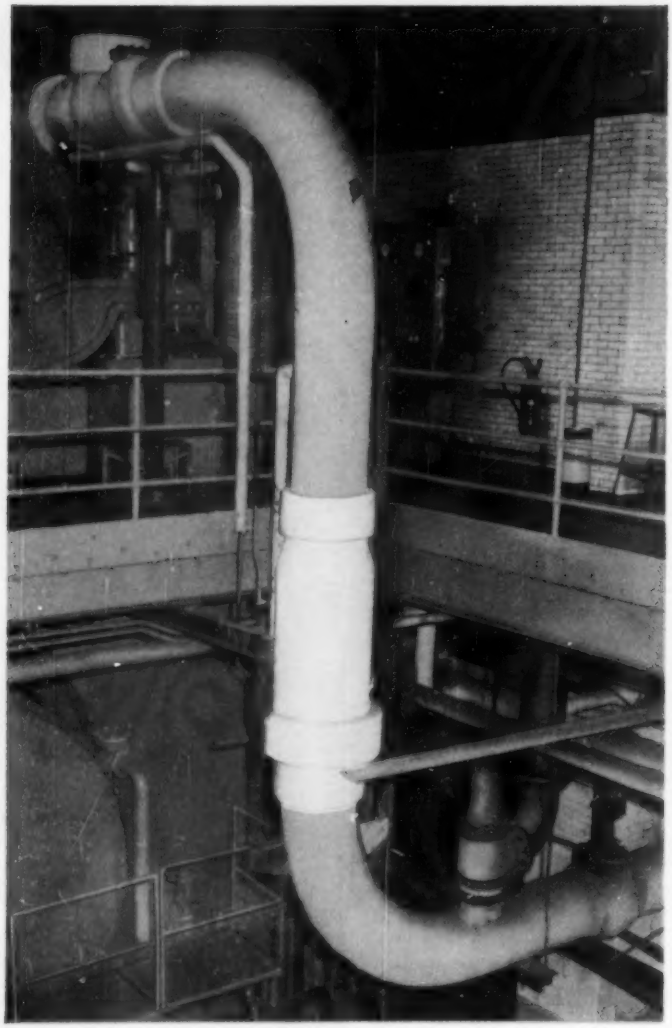
M KELLOGG W

New Copes-Vulcan Desuperheater handles critical job at Pennsylvania Electric Company



Copes-Vulcan's newly developed Variable-Orifice Desuperheater provides precise temperature control at Pennsylvania Electric Company's Front Street Station. Installed in the steam header supplying a 12,500-kw turbine generator, this desuperheater has maintained a discharge steam temperature of 490 FFT even though inlet temperature has varied between 600° and 650°.

Located just 20 feet upstream of the turbine throttle this desuperheater holds temperatures within a plus or minus 3F limits, consistently meets all required pressure conditions.

Advance design pays off in other ways, too. Only one outside connection —for cooling water—is needed. With only one moving part, this desuperheater presents no maintenance problem. No long runs of piping are needed to mix fluids. No atomizing steam, spray nozzle or glands are required.

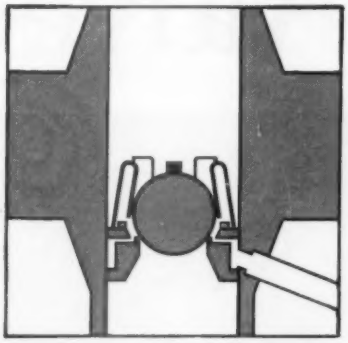
Copes-Vulcan designs a complete line of desuperheaters, each type engineered to meet particular operating conditions. For complete details on the Variable-Orifice Desuperheater, write for Bulletin 1037.

COPES VULCAN

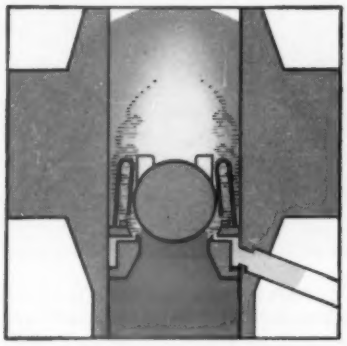
Copes-Vulcan Division
BLAW-KNOX COMPANY
Erie 4, Pennsylvania

BLAW-KNOX

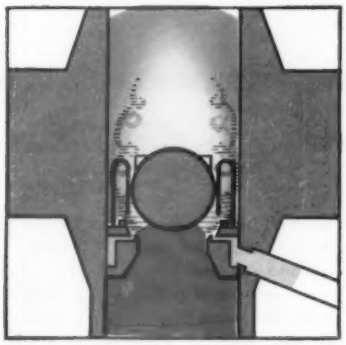
SIMPLE PRINCIPLE OF OPERATION



No load: Ball has seated itself on the ring. There is no flow of steam—no flow of cooling water.

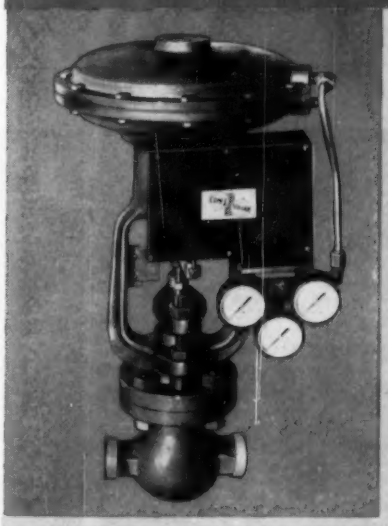


Light load: Ball opens orifice slightly. Cooling water is instantly and intimately mixed with steam.



Full load: Orifice fully opened. There is the same pressure drop and high turbulence at all loads over the full range.

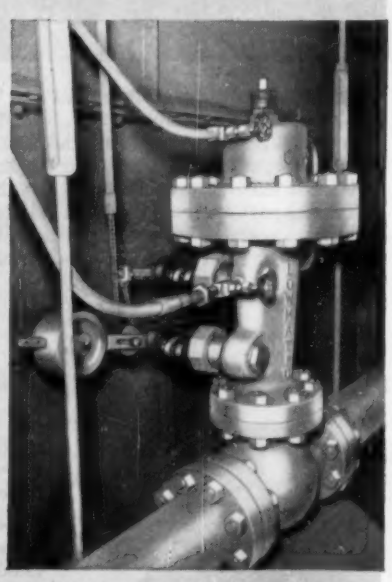
C-V NEWS NOTES



A Type CV-D Pressure reducing valve is used at Front Street Station to automatically regulate the cooling water supplied from the feed-water-heater.

This Copes-Vulcan diaphragm valve is designed to dependably meet rigid standards of accuracy for pressure, temperature and liquid level control. The Type CV-D may be direct or reverse acting. Rangeability is excellent.

Style and area of valve ports are scientificaliy selected to meet specified operating conditions. These versatile valves can be applied in sizes up to 12 inches in pressure—standards up to 1500 pounds. For details on Copes-Vulcan valves, write for Bulletin 1027.



For modern feed water control: The Copes Flowmatic, a two-element feed water regulator, is sensitive to the slightest change in steaming rate, and to the most minute fluctuation of boiler water level.

Closer stabilization of boiler water level under all load conditions makes the Copes Flowmatic ideal for the modern utility and industrial boilers subject to rapid, wide fluctuations in steam demands.

With the direct-operated Flowmatic Regulator (shown above) the steam flow element is built integral with the valve bonnet. In regular service outdoors, the Flowmatic Type D-O is unaffected by atmospheric conditions, provides accurate, dependable performance. For further information, write for Bulletin 1003-A.

Here's what makes Bronze Valves

TYPICAL OF WALWORTH QUALITY is the union body-to-bonnet connection which stiffens the body against internal pressure; makes taking the valve apart a simple operation and reduces the chances of distortion or leakage even though the valve is repeatedly taken apart and reassembled. With this type of construction there is no possibility of the bonnet coming off the valve while the handwheel is being turned.

HEAVY BODY CONSTRUCTION is typical of *all* Walworth Bronze Valves. Extra-thick walls and rugged wrench hexes constitute a high safety factor and prevent distortion while the valve is being installed in the pipeline. Extra-deep pipe threads are accurately machined to eliminate leakage. Walworth Bronze Valves are also available with flanged, silver-brazed or soldered ends in certain sizes and types.

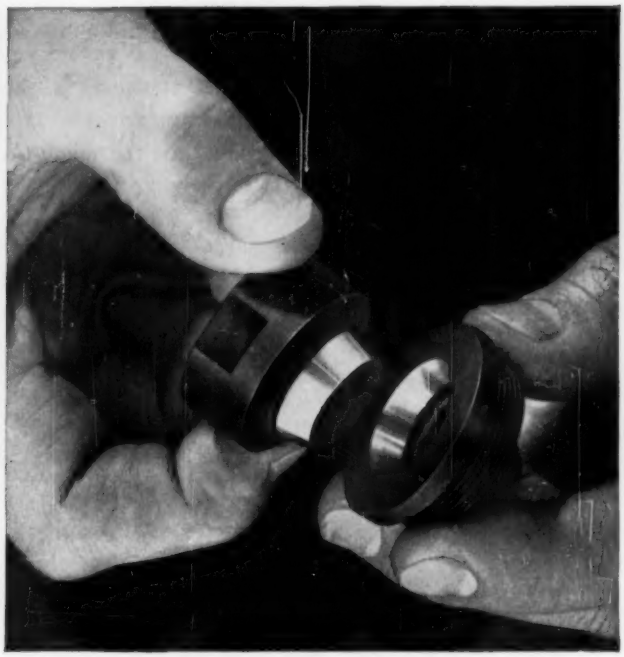
MEET THE CHAMP! The chief engineer of a midwest plant had used a parade of valves in severe boiler blowdown service. Most didn't last longer than 60 days. None survived 90 days.

On the recommendation of a Walworth Representative the engineer installed the Walworth No. 225P Bronze Globe Valve shown here, stating that he would be entirely satisfied if it lasted a bare three months. Exactly 4 years and 362 days later the valve was taken out of service—not due to the wire drawing, steam cutting and galling which made the other valves short-lived—but because the highly turbulent steam finally wore a small hole in the body.

This is the kind of valve satisfaction you get—when you specify and use Walworth Bronze Valves. They are the longest wearing, toughest bronze valves on the market.

Walworth Subsidiaries: ALLOY STEEL PRODUCTS CO. • CONOFLOW CORPORATION • GROVE VALVE AND REGULATOR CO.

Walworth *the* real bargain!



EXTRA-LARGE STEMS with extra-long, extra-deep threads prolong valve life, protect against wear and distortion and provide tight positive shutoff. The surface of the stem is machined to a glass-like finish for minimum handwheel effort and to preserve the packing which results in fewer inspections and less maintenance. The top of the stem is tapered and squared to hold the handwheel securely.

TO REDUCE WIRE DRAWING to a minimum, certain types of bronze globe valves have stainless-steel plug-type seats and discs heat-treated to a nominal hardness of 500 Brinell, adding years to valve life even in severe services. These valves can be tightly closed on sand, grit or pipe scale without damage. Seats and discs are machined simultaneously, assuring perfect mating.

There is a Walworth Bronze Gate, Globe, Angle or Check Valve for every service. Walworth is continually developing new valve types and materials, including plastics, to keep pace with the growing variety and severity of services in modern industry. For full information, see your Walworth Distributor or write:
Walworth, 750 Third Avenue, New York 17, N. Y.

WALWORTH

Bronze Valves and Fittings

M & H VALVE & FITTINGS CO. • SOUTHWEST FABRICATING AND WELDING CO., INC. • WALWORTH COMPANY OF CANADA, LTD.

The trend to BIG

It is widely recognized that in recent years there has been a very definite trend to "BIG" units. In the early 'fifties, a number of orders were placed for units in the 200 mw range, and, by 1952, the maximum size had reached 250 mw. Since then, a substantial number of units for capacities from 250 mw up have been ordered.

The following paragraphs reveal the extent to which C-E Boilers have been chosen for this "250 mw and up" category and give some interesting highlights of these large units:

- Nineteen C-E Boilers have been ordered thus far to serve turbine-generators with capacities from 250 mw up. The largest—rated at 500 mw—is the biggest unit ordered to date. The aggregate capacity of these units is well over 5½-million kilowatts.
- All but two of the C-E Boilers in this size range are of the controlled circulation design, and these two are C-E Sulzer Monotube Boilers for supercritical pressures.
- All but one of the boilers incorporate the C-E Twin-Furnace design . . . service-proved as the "Best for Big Boilers."
- All the boilers are designed to supply steam at *throttle pressures* from 2,000 psig up. The two C-E Sulzer units are to operate at supercritical pressure . . . one at 5,000 psi.
- The size pattern of these boilers may have some significance. There are 6 of 325 mw capacity, 5 of 275 mw, 4 of 250 mw and one each of 260, 300, 340 and 500 mw. The average is over 300 mw.

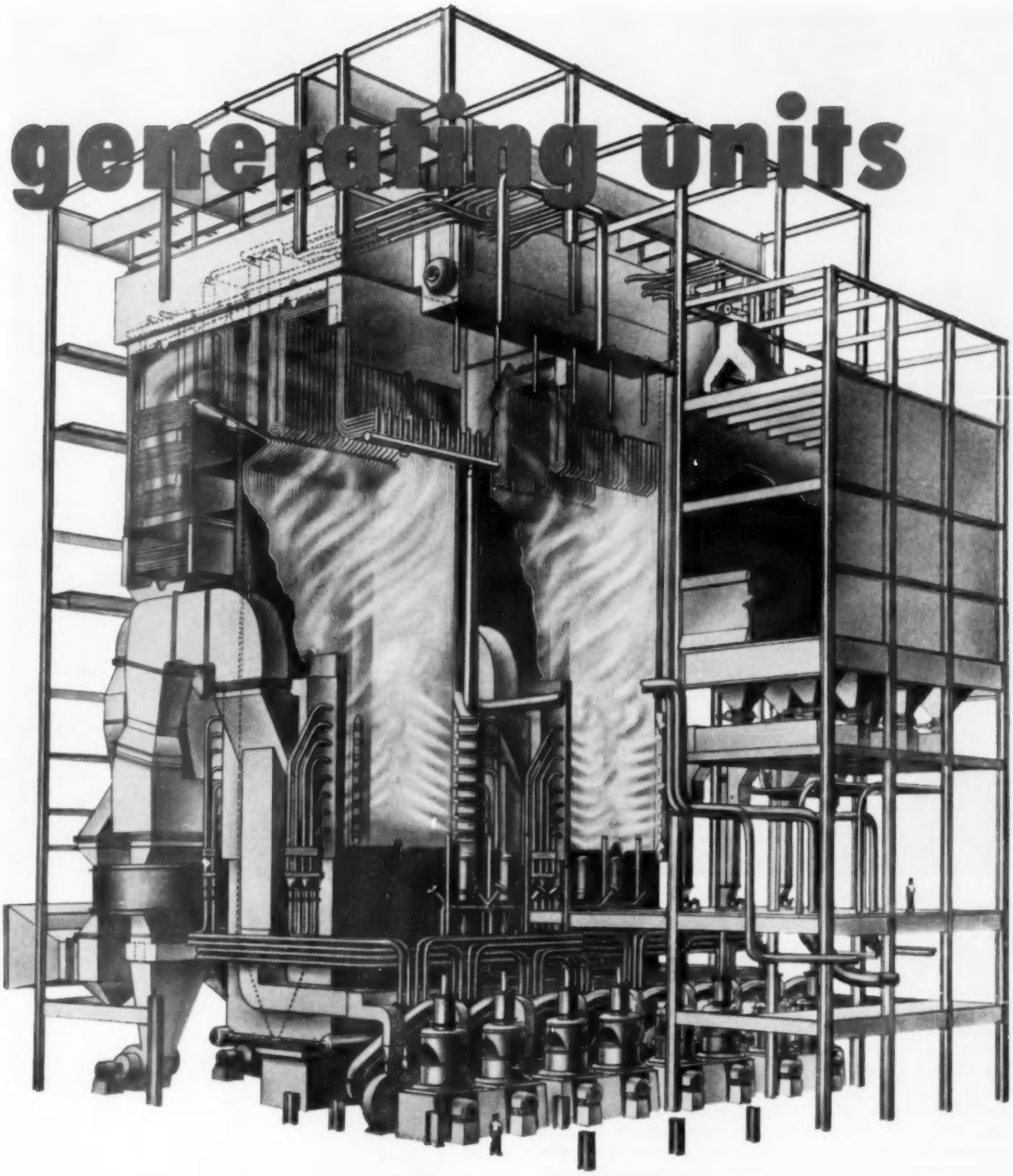
The extent to which C-E Boilers have been selected in this important size area evidences the widespread recognition of such design advantages as Controlled Circulation, Twin-Furnace arrangement and Tangential Firing. These features of C-E "Big" Boilers offer greater design flexibility, faster load response, steadier water level, shorter outages for maintenance and quicker start-ups, as well as tight control over superheat and reheat temperatures and minimum use of spray water for desuperheating. They add up to important plus values in today's "Big" Boiler field.

COMBUSTION

Combustion Engineering Building

ALL TYPES OF STEAM GENERATING, FUEL BURNING AND RELATED EQUIPMENT; NUCLEAR REACTORS;

generating units

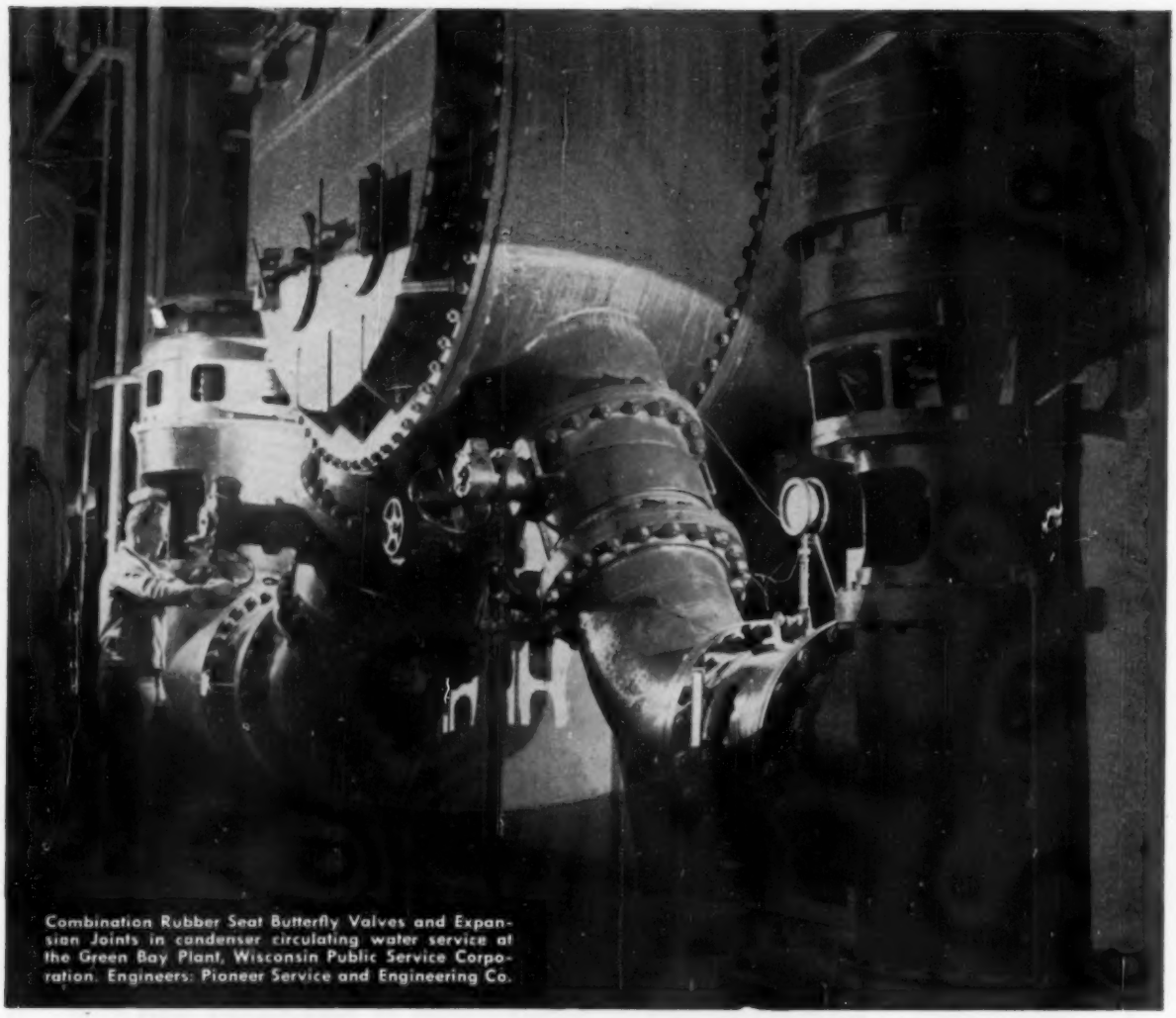


THE C-E CONTROLLED CIRCULATION TWIN-FURNACE BOILER

ENGINEERING C-150

200 Madison Avenue, New York 16, N. Y.

PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE



Combination Rubber Seat Butterfly Valves and Expansion Joints in condenser circulating water service at the Green Bay Plant, Wisconsin Public Service Corporation. Engineers: Pioneer Service and Engineering Co.

GREEN BAY...Combines Butterfly Valves with expansion joints for new economy

Drop-tight shutoff, plus provision for pipeline expansion and ease of assembly are provided here in one compact Pratt unit. By combining the Butterfly Valve with the expansion joint, space requirements are minimized and maintenance (and initial cost) of one set of flanges and bolts is eliminated. This is a typical Henry Pratt engineering job—the mechanisms are the simplest available . . . for economy and minimum maintenance, and they are carefully designed and built for peak efficiency and operating ease.

Henry Pratt pioneered the use of Rubber Seat Butterfly Valves in power plants. Combined with permanently drop-tight shutoff, the inherent simplicity and compactness of this valve permitted a new concept of large-diameter valving by power plant engineers.

Pratt Rubber Seat Butterfly Valves grew with the Power Industry for thirty years, and today are being installed in modern, nuclear power plants. For valve design—with **imagination**—see Henry Pratt.

NEW! Latest, most accurate pressure drop and flow data, conversion tables, discussion of butterfly valve theory and application plus other technical information . . . Write for ManualB-2D.

HENRY PRATT

RUBBER SEAT Butterfly Valves

Henry Pratt Company, 2222 S. Halsted St., Chicago 8, Ill. Representatives in principal cities

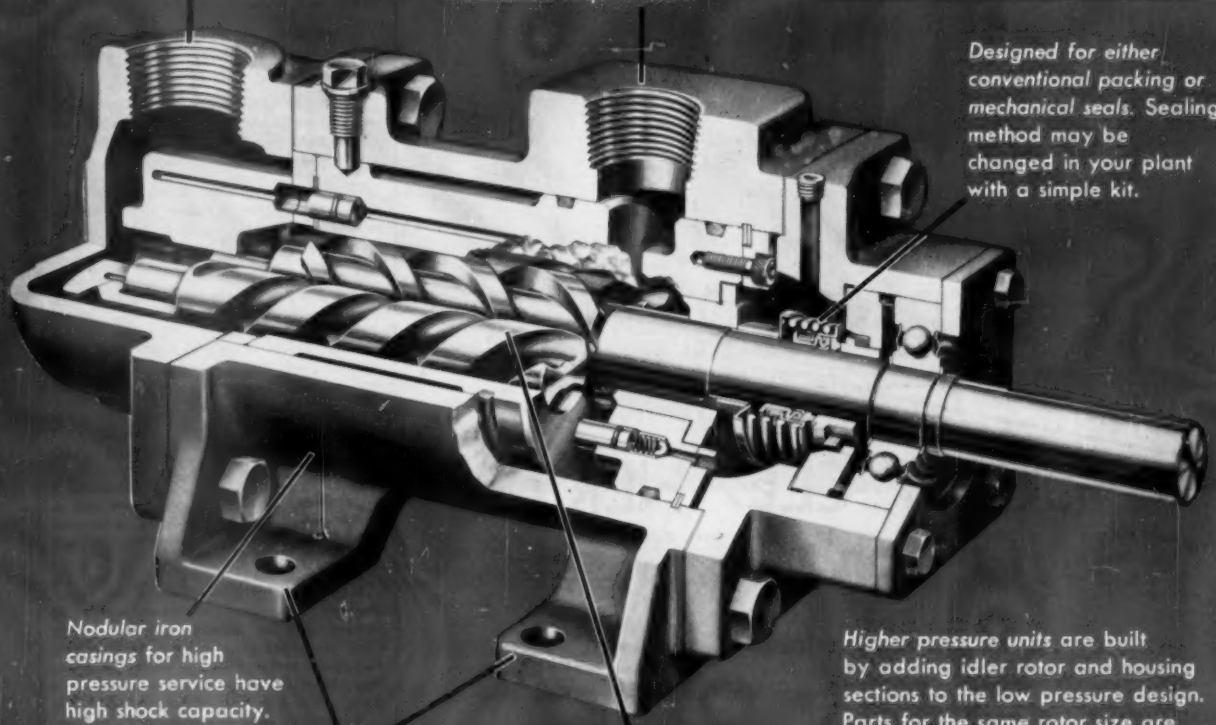
DE LAVAL
IMO PUMPS

are now more versatile than ever

De Laval IMO pumps have proved that they do a dependable job over long years of service. The reason is IMO design simplicity. These constant displacement rotary pumps have only three moving parts—smoothly intermeshing rotors that propel the fluid axially in a steady flow without churning, pocketing or pulsation. There are no timing gears, cams, valves, sliding vanes, or reciprocating parts to wear or become noisy. *Quiet*, compact IMO pumps are excellent for direct-connected, high-speed operation.

In addition to these basic pumping advantages, the improved IMO gives you important new benefits shown in the cutaway illustration below.

DE LAVAL IMO PUMPS can also be used as hydraulic motors.



Inlet can be rotated to suit installation arrangement.

Discharge flanges are infinitely varied. You can use the most advantageous piping method to suit installation requirements.

Designed for either conventional packing or mechanical seals. Sealing method may be changed in your plant with a simple kit.

Nodular iron casings for high pressure service have high shock capacity.

Any position mounting is possible without factory modification.

Higher pressure units are built by adding idler rotor and housing sections to the low pressure design. Parts for the same rotor size are interchangeable over the entire pressure range.

Internal parts are designed as a package so that units can be built into your machines.

Bulletin 3001 gives data on improved De Laval IMO pumps. Send for your copy today.

DE LAVAL *IMO Pumps*

DE LAVAL STEAM TURBINE COMPANY

886 Nottingham Way, Trenton 2, New Jersey

DL 418

THE BAYER CO.

MANUFACTURERS OF BAYER SOOT BLOWERS

For Highest First and Final Value

. . . BUY BAYER!

BAYER

Balanced Valve

SOOT CLEANER

Bayer's single-chain design compels perfect in-step operation of valve and element. Operation is positive, definite, assuring a full flow of steam for efficient cleaning.

When the operator pulls chain, the cam-actuated, quick-action balanced valve is opened. By continued pulling of the chain, worm drive slowly rotates element over cleaning arc. When element reaches end of cleaning arc, valve automatically closes.

Minimum steam consumption—low maintenance. Every detail is engineered, built for long life, efficient performance at high temperatures and high pressures.

More than 30,000 boilers are Bayer equipped. More than 45 years' successful, specialized experience assures you investment economies in Bayer equipment.

QUALIFIED LOCAL ENGINEERING SERVICE—Your Bayer representative is an experienced engineer, well qualified to take care of any service needs in connection with Bayer Soot Cleaners. He is available when you call upon him and will gladly render any service required.

All These Mechanical and Operating Advantages are available in

The BAYER

Balanced Valve

SOOT CLEANER

1. **Sound engineering, workmanship, and materials of the best.**
2. **An organization of over 45 years' experience, capable and willing to render service at all times.**

SINGLE CHAIN: Valve and element controlled by a single chain.

VALVE BODY: Rugged construction, built to last. Short and ample steam passage giving very small pressure drop.

ORIFICE PLATE VALVE: For high pressure service, each head may be controlled by an orifice plate valve through which pressure is adjusted for each individual element.

STUFFING BOX: Due to maintenance of perfect alignment on swivel tube, packing needs little attention. Stuffing box is in full view, readily accessible.

AIR SEAL: Has machined surface on wall sleeve and spring-held floating seal to prevent air in-leakage.

HEAD BEARINGS: There are two main bearings, an outboard and an inboard bearing for the swivel tube to maintain alignment.

THRUST BEARING: Ring type thrust bearing takes the load.

VACUUM BREAKERS: Two vacuum breaker air valves, or one valve and a signal whistle above each valve, to prevent suction of boiler gases into valve and piping.

ELEMENT OPERATION: With the Bayer element operation, balanced valve is opened just as element rotates, giving FULL pressure over entire cleaning arc. Full steam pressure insures thorough cleaning. Balanced valve saves wear of valve parts. With any type of poppet valve, this is important...ask any operator.

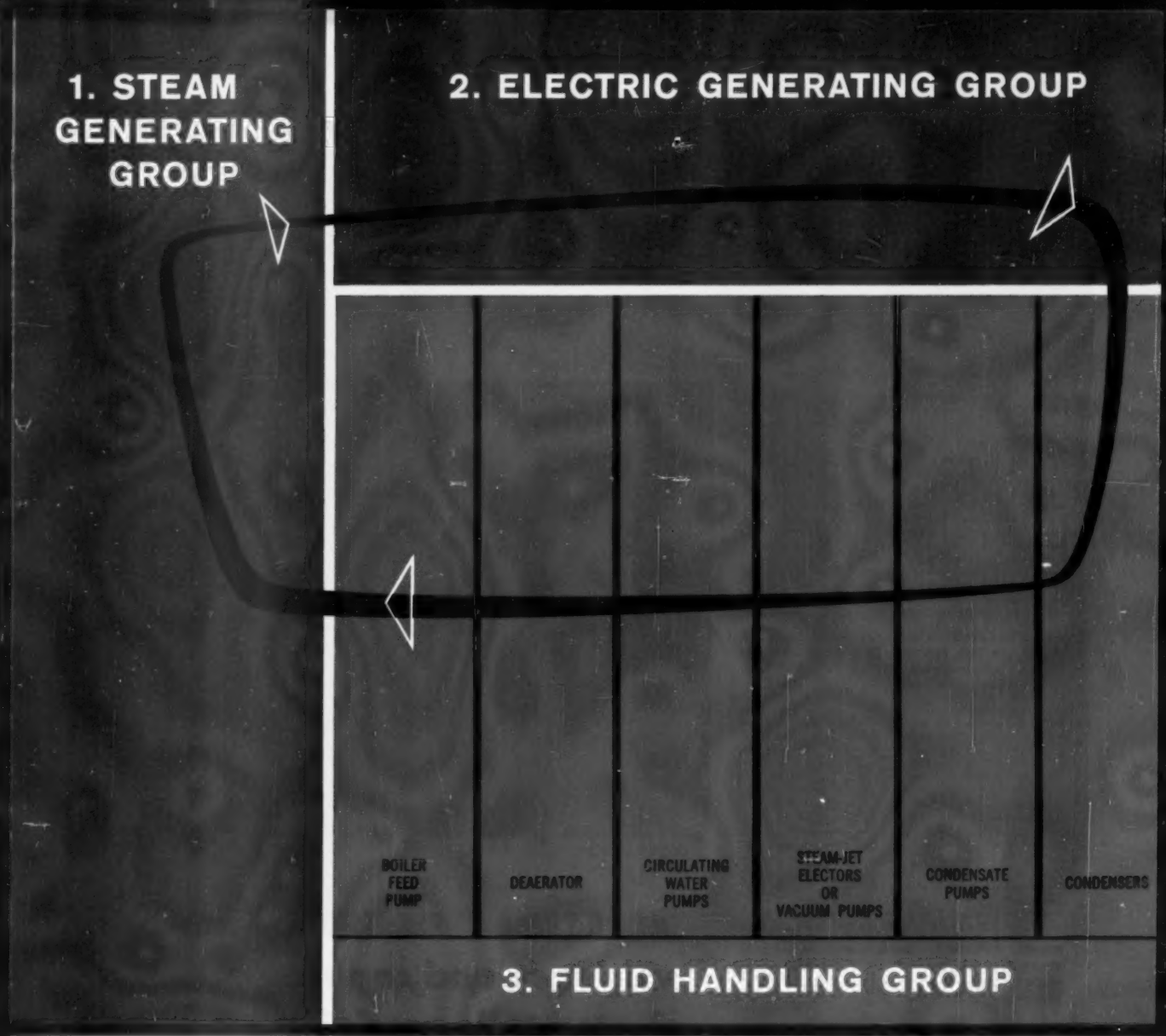
BLOWING ARC: Valve cams automatically regulate cleaning arc.

REDUCTION GEARS: 24 to 1 gear ratio gives slow rotation for good cleaning.

FLANGED PIPE CONNECTION: Operating head is connected to supply pipe by flanges and through bolts, or high tensile studs and nuts.

THE BAYER CO.

St. Louis, Mo.

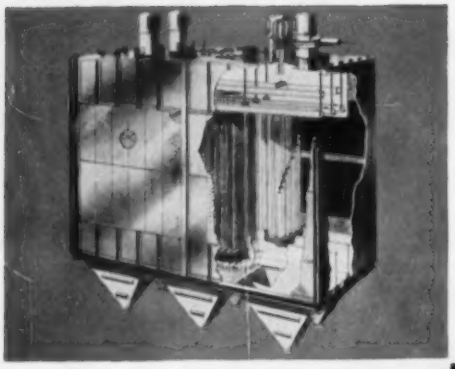


LOOK AT ALL THREE FOR POWER

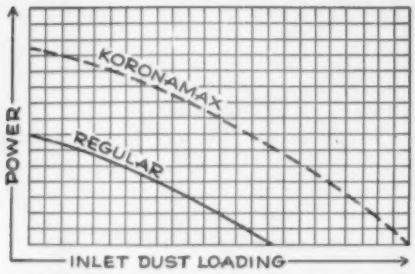
THE FLUID HANDLING GROUP in a steam power plant has as many individual components as the other two combined. Failure of any one component can disrupt operation of the whole. Reliability can best be achieved through compatible integration of the wide range of equipment in this all-important group. As a manufacturer of all major fluid handling components, Worthington's "system-wise" experience and knowledge can be of extreme value to you. For information, write or call your nearest Worthington district office. Worthington Corporation, Harrison, New Jersey.

49-1

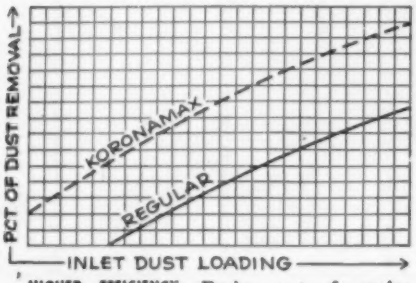
WORTHINGTON



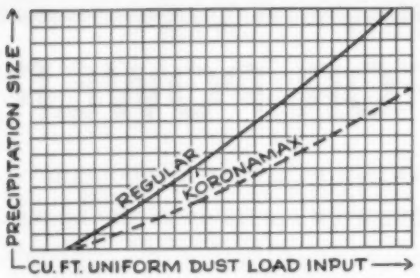
Another Koppers Exclusive in ELECTROSTATIC PRECIPITATION



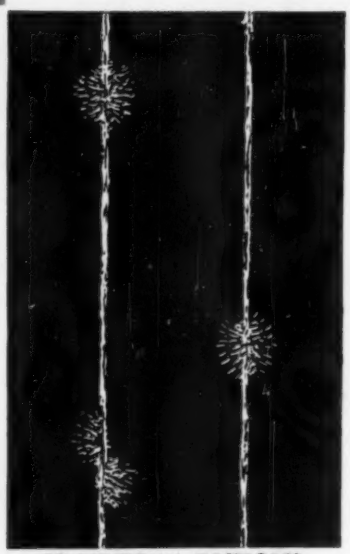
INCREASED POWER—The evenly spaced discharge points of "Koronamax" Electrodes reduce the arc-over tendency and permit increasing power input.



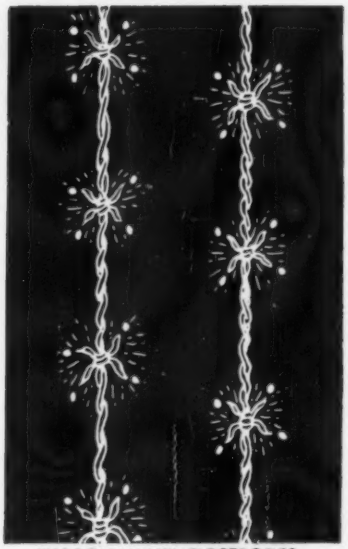
HIGHER EFFICIENCY—Replacement of regular electrodes with "Koronamax" Electrodes allows higher power input — greatly increased efficiency.



SMALLER SIZE—In new installations desired efficiency may be obtained with smaller unit when "Koronamax" Electrodes are used.



CONVENTIONAL ELECTRODES



"KORONAMAX" ELECTRODES

"KORONAMAX" ELECTRODES increase efficiency and capacity of electrostatic precipitators

"Koronamax" Electrodes developed by Koppers are now in service in several different applications and their *controlled* corona discharge has resulted in greatly increased capacity and efficiency. This unique type of precipitator electrode may solve *your* gas cleaning problem.

Koppers' experience, constant research and extensive field testing have led to this important advance in precipitator design. Check with Koppers to see if "Koronamax" Electrodes can help you get top precipitator efficiency. For more information, write: KOPPERS COMPANY, INC., 4405 Scott Street, Baltimore 3, Md.

KOPPERS

ELECTROSTATIC PRECIPITATORS

Engineered Products Sold with *Service*

Cities Service shows how to cut valve costs

Crane globe valve in 14th year on steam turbine line

Cities Service Refining Corp. uses hundreds of valves at its Lake Charles, La., plant. Maintenance and replacement costs could be staggering, but for careful control of valve selection.

For example, the Crane 6-inch, 300-pound steel globe valve shown above was installed at the Lake Charles plant 14 years ago. Working at 250 psi., 600 deg. F., the valve is used in the steam line to a turbine-driven feed-water pump.

In spite of its operation 3 or 4 times a week, this sturdy Crane valve has continued tight in service and has required no repairs in 14 years!

Thrifty buyers want no better reward than the kind of cost-cutting, dependable service they get when Crane valves are installed.

That's why in the petroleum industry, as in all industries, you'll find more Crane valves used than any other make.

This is the valve used on installation above. CRANE 151XR, 300-pound cast steel globe valve for 850° F. maximum temp. Sizes: 2 in. to 8 in. For full details, see your Crane Representative, or write to address below.

CRANE VALVES & FITTINGS

PIPE • PLUMBING • KITCHENS • HEATING • AIR CONDITIONING

Since 1855—Crane Co., General Offices: Chicago 5, Ill. Branches and Wholesalers Serving All Areas

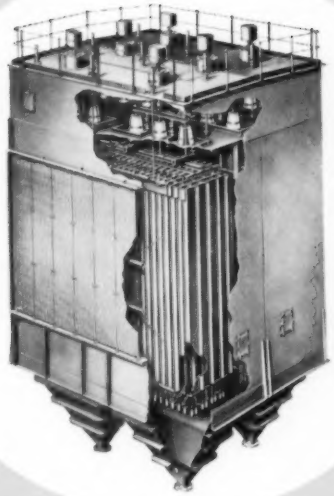
Research-Cottrell

makes all three

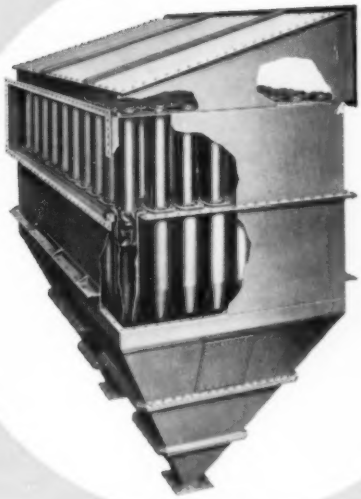
All three Research-Cottrell dust collectors are engineered for specific fly ash applications. Research's experience as the world's largest manufacturer of precipitators is well known. Over 600 straight and combination fly ash precipitators have been installed throughout the United States and Canada. • Research's Cyclo-trell, available since 1956, has already set new standards of efficiency for mechanical dust collectors. • For more information on Research-Cottrell's line of engineered dust collecting equipment, write for Bulletin PC.

Research-Cottrell

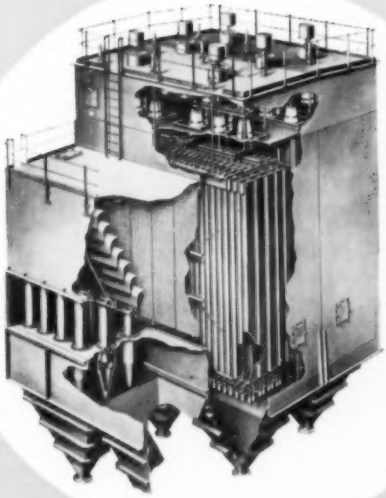
RESEARCH-COTTRELL, INC. Main Office and Plant: Bound Brook, N. J. • 405 Lexington Ave., New York 17, N. Y. • Grant Building, Pittsburgh 19, Pa. • 228 N. La Salle St., Chicago 1, Ill. • 58 Sutter Street, San Francisco 4, Calif. • Research-Cottrell (Canada) Ltd., 33 Bloor Street East, Toronto 5, Ontario.



Cottrell
Precipitators



Cyclo-trell
Mechanical Collectors



Combination
Electrical-Mechanical
Collectors

FUEL SATISFACTION IS THE No. 1 CHOICE OF MEN WHO KNOW COAL

FUEL SATISFACTION IS NATURALLY SUPERIOR
It comes from deeper, older seams; has been "refined" longer and better by Mother Nature.

FUEL SATISFACTION IS PROCESSED IN MODERN COAL "PLANTS"
It is cleaned and graded — virtually manufactured — to meet the exacting and varying needs of industry.

FUEL SATISFACTION IS TRANSPORTED DEPENDABLY
For more than 70 years, the N&W has hauled *Fuel Satisfaction* from modern mines to major markets.

The superior all-purpose Bituminous Coal mined along the Norfolk and Western is called *Fuel Satisfaction* because it satisfies every industrial fuel requirement.

It is nature's finest coal, unmatched for cleanliness and purity; processed to tailor-made perfection; dependably transported by the Norfolk and Western.

For heat, power or special purpose — for foremost performance on any job — you can't buy better coal than *Fuel Satisfaction.* It's the *No. 1 choice* of men who know coal.

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Roanoke, Virginia

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833 Chamber of Commerce Building
Telephone Liberty 2-2229
Boston 10, Massachusetts

CHICAGO
Room 604,
208 South LaSalle Street
Telephone RAndolph 6-4634
Chicago 4, Illinois

CINCINNATI
913 Dixie Terminal Building
Telephone DUnbar 1-1325
Cincinnati 2, Ohio

CLEVELAND
Room 722,
The Illuminating Building
Telephone MAin 1-7960
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1907 Book Building
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WOodward 1-2340 or 1-2341
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ST. LOUIS
2059 Railway Exchange Building
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WINSTON-SALEM
1105 Reynolds Building
Telephone PArk 2-7116
Winston-Salem 1, North Carolina

Norfolk and Western RAILWAY

CARRIER OF FUEL SATISFACTION

Unretouched photo shows what happens when a boiler feed line is clogged with calcium carbonate deposits.
CAUSE: lack of chemical stability in feed water.
REMEDY: properly balanced water treatment.

BUILD-UP FOR A BREAKDOWN

This heavy build-up of scale is not far from completely blocking the carrying capacity of this pipe. When the inevitable breakdown occurs, expensive maintenance and down time result.

You can save money by preventing costly breakdowns like this. The answer is proper water treatment wherever corrosive elements threaten.

A Consulting Service for You. Whatever your water problems...analysis, testing or treatment procedures for boiler water, condensate, cooling towers, process waters...Dearborn has the engineers, the experience and the laboratory facilities to assist you in developing the program exactly suited to your needs. Your Dearborn Engineer will outline the many advantages a Dearborn Consulting Service Program will give you.

WHY NOT CALL HIM IN... **MEANWHILE, MAIL THE COUPON FOR YOUR COPY OF THE DEARBORN CONSULTING SERVICE BOOKLET**

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Manufacturing plants in CHICAGO • LINDEN • LOS ANGELES • TORONTO • HONOLULU • HAVANA • BUENOS AIRES

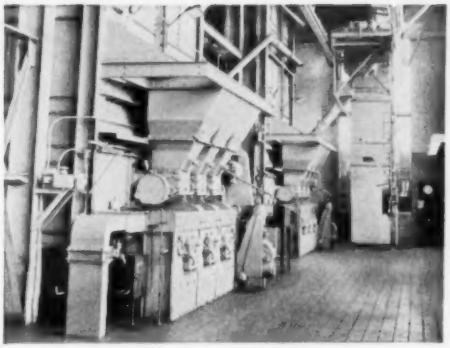
Coal scores

with Notre Dame

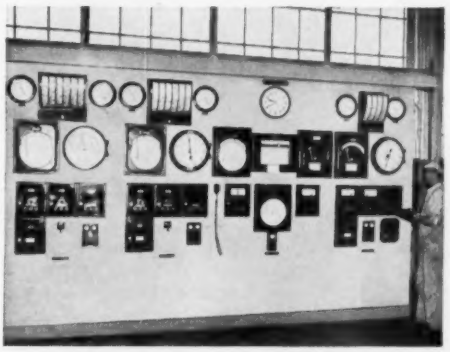
University power plant burns coal for modern steam generation

Enrollment growth and building expansion had put a strain on the power plant at the University of Notre Dame, South Bend, Ind. After careful study—by the consulting firm of Albert Kahn Associated Architects and Engineers, of Detroit—a decision was made to modernize steam facilities and add power generating equipment. Two new coal-fired boilers, a turbo-generator and auxiliary equipment were installed.

Maximum efficiency is achieved through automatic combustion control and complete instrumentation. Today steam is generated *economically.* Another noteworthy result of these innovations has been the *cleanliness* of operation . . . making Notre Dame's power plant a model of good housekeeping.



Firing aisle shows both Union Iron Works boilers . . . identical 2-drum, bent-tube type with a normal rating of 70,000 lb/hr. These units are fired by Detroit Rotograte Spreader Stokers.



All combustion controls and instruments are centered in this control panel by Hays Corp.

Furnace ash and dust from Prat-Daniel Collectors are conveyed pneumatically to this 50-ton ash storage silo. Rotary dustless unloader assures clean removal of material. Ash-handling system is by United Conveyor Corp.

Facts you should know about coal

You'll find that bituminous coal is not only the lowest-cost fuel in most industrial areas but up-to-date coal burning equipment can give you 15% to 50% more steam per dollar. Today's automatic equipment can pare labor costs and eliminate smoke problems. And vast coal reserves plus mechanized production methods mean a constantly plentiful supply of coal at stable prices.

Technical advisory service

To help you with industrial fuel problems the Bituminous Coal Institute offers a free technical advisory service. We welcome the opportunity to work with you, your consulting engineers and architects. If you are concerned with steam costs, write to the address below. Or send coupon below for our case history booklet, complete with data sheets. You'll find it informative.

Consult an engineering firm

If you are remodeling or building new heating or power facilities, it will pay you to consult a qualified engineering firm. Such concerns—familiar with the latest in fuel costs and equipment—can effect great savings for you in efficiency and fuel economy over the years.

BITUMINOUS COAL INSTITUTE

Department C-05

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SEND COUPON FOR YOUR COPY OF "Guide Specifications for Typical Low-Pressure Commercial Heating Plant." This booklet, adaptable for design loads 3,000 to 24,000 EDR steam, contains specifications, drawings and tables on all aspects of a typical heating plant. C-05

☐ Guide Specifications Booklet ☐ Case histories on larger plants

Name _____

Title _____

Company _____

Address _____

City _____ Zone ____ State _____

POWELL

world's largest family of valves

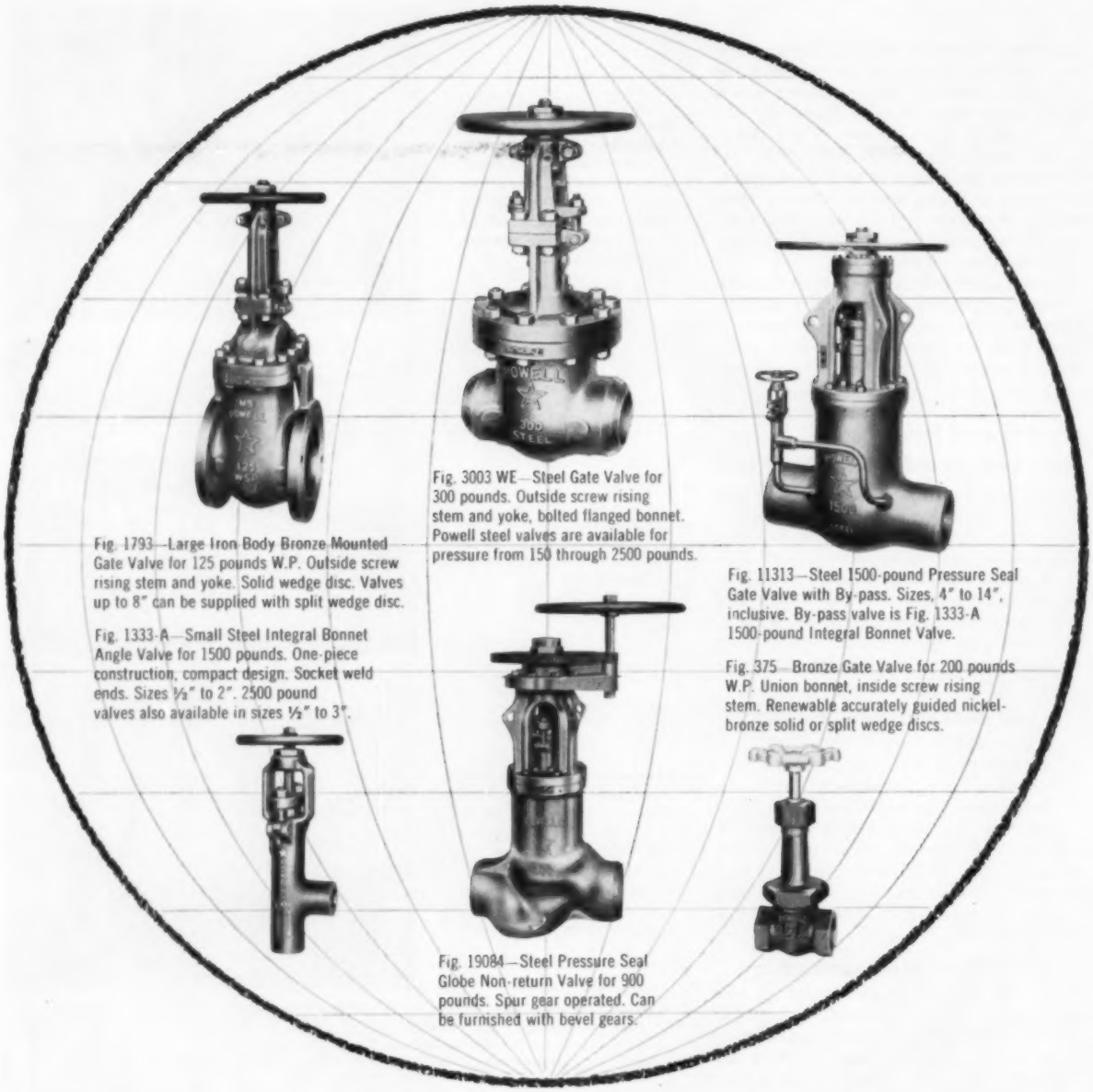


Fig. 1793—Large Iron Body Bronze Mounted Gate Valve for 125 pounds W.P. Outside screw rising stem and yoke. Solid wedge disc. Valves up to 8" can be supplied with split wedge disc.

Fig. 1333-A—Small Steel Integral Bonnet Angle Valve for 1500 pounds. One-piece construction, compact design. Socket weld ends. Sizes ½" to 2". 2500 pound valves also available in sizes ½" to 3".

Fig. 3003 WE—Steel Gate Valve for 300 pounds. Outside screw rising stem and yoke, bolted flanged bonnet. Powell steel valves are available for pressure from 150 through 2500 pounds.

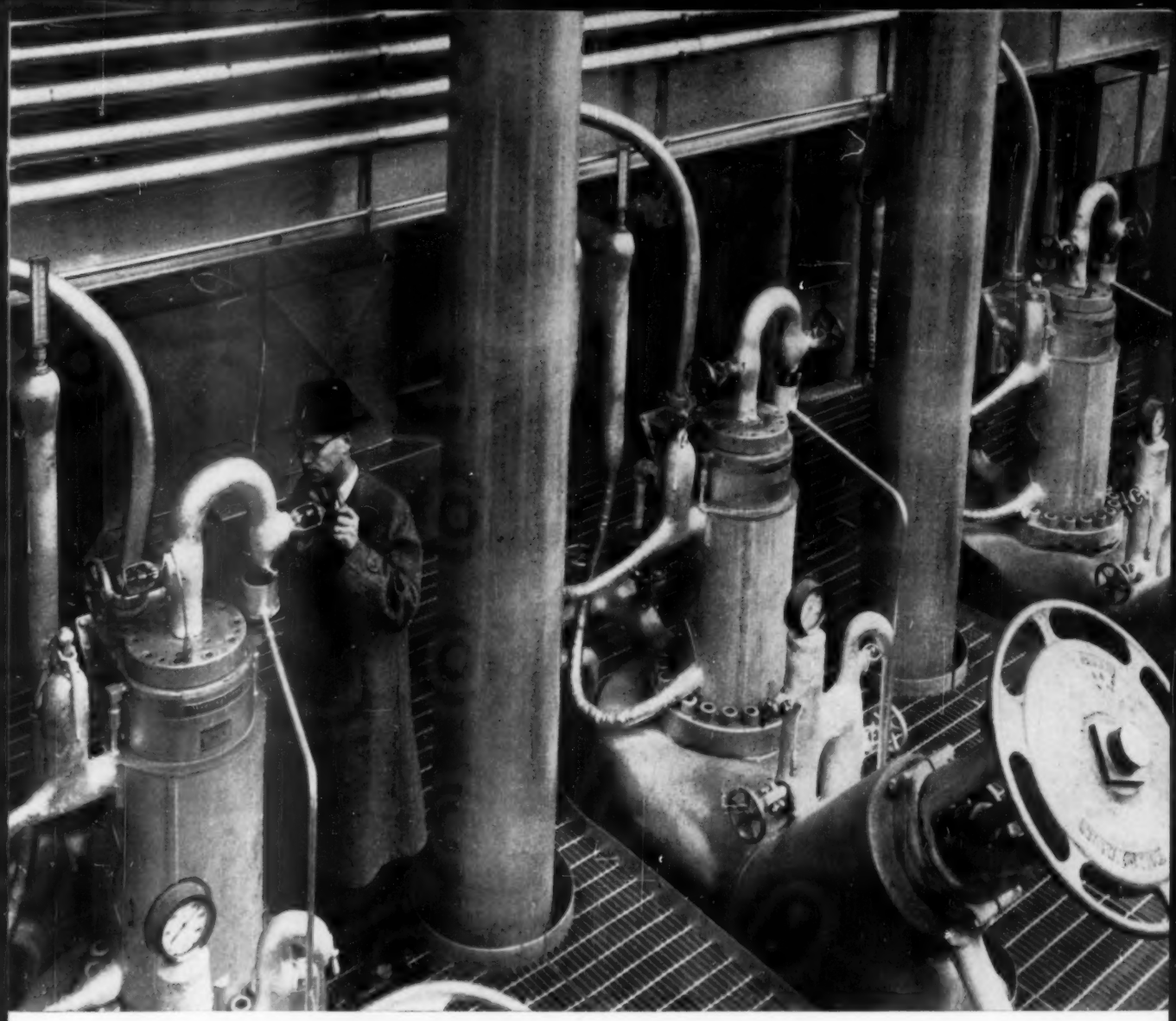
Fig. 19084—Steel Pressure Seal Globe Non-return Valve for 900 pounds. Spur gear operated. Can be furnished with bevel gears.

Fig. 11313—Steel 1500-pound Pressure Seal Gate Valve with By-pass. Sizes, 4" to 14", inclusive. By-pass valve is Fig. 1333-A 1500-pound Integral Bonnet Valve.

Fig. 375—Bronze Gate Valve for 200 pounds W.P. Union bonnet, inside screw rising stem. Renewable accurately guided nickel-bronze solid or split wedge discs.

A solution for every kind of flow control problem is as near as your local Powell distributor. Powell valves are designed and engineered in the largest variety of metals and alloys, to handle any medium, every flow control requirement. There are Powell distributors in all principal cities. Or, if yours is a special engineering problem, write to:

THE WM. POWELL COMPANY • *Dependable Valves Since 1846* • **Cincinnati 22, Ohio**



Westinghouse canned motor-pumps

specified for KP&L outdoor installation

The Kansas Power and Light Company recently increased the capacity of its Tecumseh Station with the addition of a 75,000-kw outdoor unit. This new addition, the most efficient in the plant, will operate day and night as the base load unit. To insure reliability, Westinghouse canned motor-pumps were specified for the Combustion Engineering Inc. controlled circulation boiler.

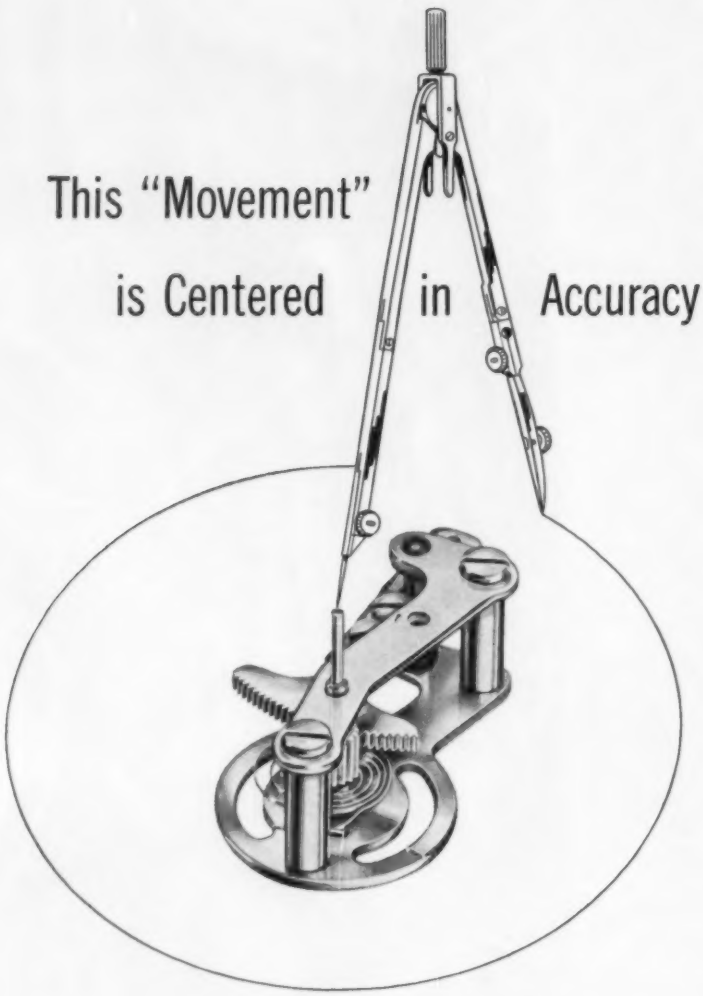
Westinghouse pumps were selected because: they have been proven for long-term continuous operation in many other installations; the zero-leakage feature eliminates injection water sealing. Two pumps provide the required boiler water circulation to deliver 635,000 pounds of 1000°F steam at 1800 psi. A third Westinghouse pump is kept on hot standby. All three are rated at 4070 gpm, 1958 psi, 615°F.

For further information on the tested and proven Westinghouse canned motor-pumps, contact your Westinghouse sales representative, or write: Westinghouse Electric Corporation, Atomic Equipment Dept., P.O. Box 217, Cheswick, Penna.

J-57019

YOU CAN BE SURE...IF IT'S Westinghouse

This "Movement" is Centered in Accuracy



The rotary movement in Ashcroft Duragauges has a geared center shaft on which the pointer is mounted. When pressure flexes the Bourdon tube, the pointer is always positively positioned. You can rely on the Duragauge for precise accuracy in measuring pressure no matter how severe the conditions of service.

Correct calibration is guaranteed: the one-piece link between movement and Bourdon tube prevents slippage or parting under tension. Recalibration is easy from front or rear. Universal adjustability permits uniformly graduated dials.

You can order Ashcroft Duragauges with all-stainless-steel movement or stainless steel with nylon bearings and pinion gear. A complete choice of Bourdon tube materials, pressure ranges, dial sizes and case designs and materials is available. Your industrial supply distributor will gladly help you select the best combination for your requirements. So, be certain of highest sustained accuracy, durability and economy – specify Ashcroft Duragauges.

Ashcroft Duragauge in Alumalife® case—a lifetime case made of special aluminum alloy.

MANNING MAXWELL & MOORE INC. M&M TRADE MARK

ASHCROFT PRESSURE GAUGES

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APEXIOR®
in today's power boilers

There's something basic about a power-field product with a job even more vital today than half a century ago — for technologically we've come a whole era since Apexior Number 1® first went to work inside boilers in 1906. The "why" of Apexior and its still-growing use in modern steam generators takes us, then, to fundamentals, best answered by asking—

WHAT IS APEXIOR?

Apexior Number 1® is a high-temperature, non-oxidizing, brush-applied coating for service on both heat-transfer and non-heat-transfer areas. Because it penetrates and becomes one with metal, Apexior seals surfaces at new, or newly cleaned, strength, isolating steel from its environment and establishing it as an independent element in steam generation.

WHY ARE BOILERS APEXIOR-COATED?

Maintaining the status quo to keep steel at peak efficiency is only the first benefit Apexior confers. The simple one-coat process actually transforms steaming surfaces. Because Apexior is smooth and deposit-repelling, metal stays cleaner longer — heat transfer is always at its highest — distribution is uniform — and circulation and evaporation proceed at top performance levels. Off the line, inspection is facilitated — cleaning, when required, is accomplished easily and quickly — and the boiler is soon back in better-than-ever steaming service.

WHERE IS APEXIOR IN USE TODAY?

Now manufactured in the United States and four foreign countries, Apexior serves the world around in electric utility central stations and plants generating power for every kind of manufacturing and service industry. Its first use half a century ago in Scotch marine boilers has been extended also over the years to floating power plants in vessels of every type and size.

CAN YOU APEXIOR-COAT ALL BOILERS?

Apexior serves under highest temperatures and pressures on any steam-or-water-contact area. It may therefore be applied to any water-tube boiler, including water walls, economizers, circulators, and reheat units — and to any fire-tube boiler where it serves on shell interiors and flue exteriors.

WHAT UNIT-SYSTEM AUXILIARIES ARE USUALLY APEXIOR-COATED?

- *Evaporators — interior surfaces, coil exteriors*
- *Deaerating heaters — shells and trays*
- *Feedwater heaters — interior surfaces*

WHAT PARTS OF STEAM TURBINES ARE APEXIOR-COATED?

- *Interior of casings and exterior of rotors, or wheels, and diaphragms.*

MAINTENANCE FOR METAL

This message is one of a continuing series. To be discussed: Apexior Number 1® in relation to . . . heat transfer, feedwater treatment, high-purity steam, standby, cleaning, cost. Watch for the answer to your question, or write

THE DAMPNEY COMPANY
HYDE PARK, BOSTON 36, MASSACHUSETTS

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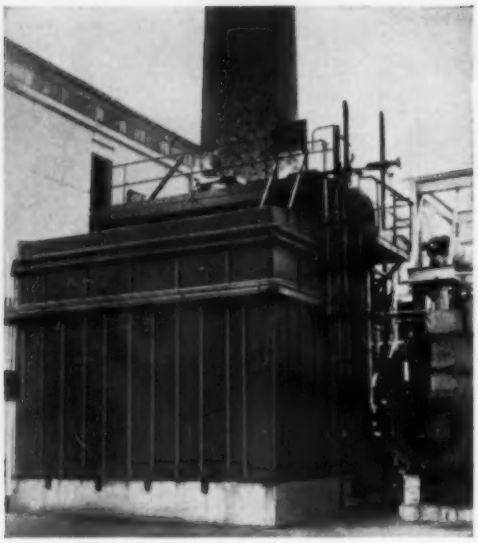
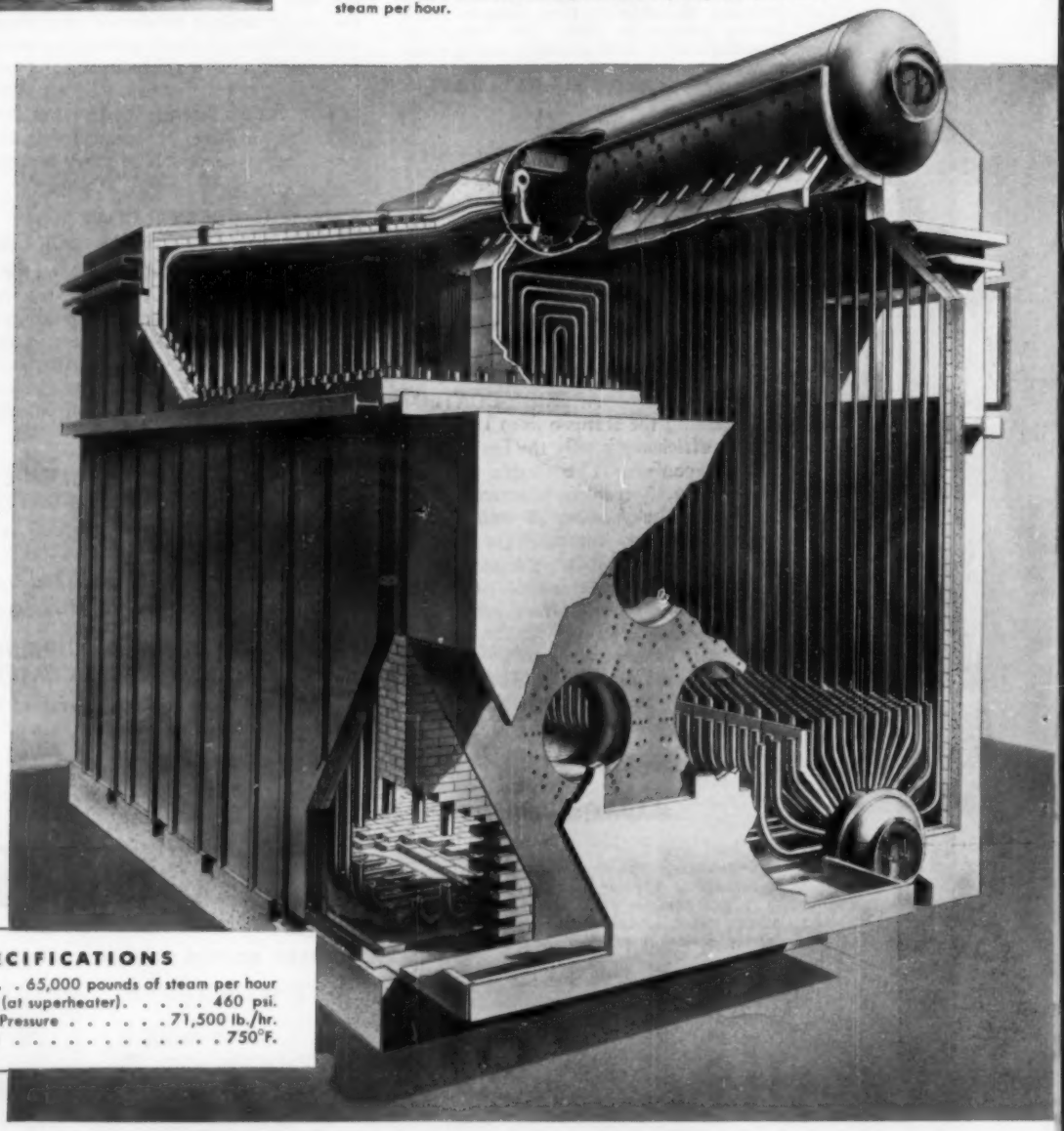


Foster Wheeler "SC" Standard at Socony Mobil Oil Company

Republic

The "SC" Series Standard Steam Generator is the latest development by Foster Wheeler Corporation toward the reduction of steam costs for industrial plants through engineering and standardization. This unit installed at the Brooklyn Refinery of Socony Mobil Oil Company, is built with Republic ELECTRUNITE Boiler Tubing. It operates continuously at its guaranteed 2-hour peak load capacity of 71,500 lbs. of steam per hour.



SPECIFICATIONS

Capacity . . . . . 65,000 pounds of steam per hour
Operating Pressure (at superheater) . . . . 460 psi.
Maximum Working Pressure . . . . . 71,500 lb./hr.
Steam Temperature . . . . . 750°F.

Boiler delivers 71,500 lbs./hr. in round-the-clock operation Brooklyn Refinery...designed and built with

ELECTRUNITE Boiler Tubing

Republic ELECTRUNITE® Boiler Tubing offers designers, erectors, and end-users built-in quality that can be measured by dependable, economical day-in, day-out service.

The "SC" Series Standard Steam Generator, designed and developed by Foster Wheeler Corporation, is providing round-the-clock power and service economies at the Brooklyn Refinery of the Socony Mobil Oil Company. This unit uses Republic ELECTRUNITE Boiler Tubes.

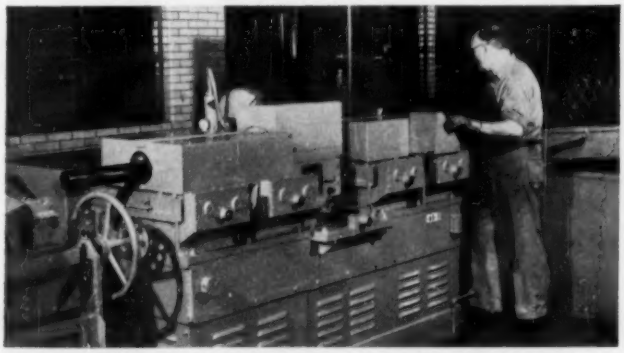
It has been found entirely feasible to operate this unit *continuously* at its guaranteed 2-hour peak load capacity of 71,500 lb. of steam per hour—6,500 lb./hr. in excess of rated capacity.

Designed for oil firing, with burners arranged for alternate burning of refinery gas, this two-drum water-tube boiler delivers 460 psi., 750°F. steam at the superheater outlet. Fitted with turbine-driven forced and induced draft fans, it is of balanced draft design, with highly efficient firing arrangement and over 19-feet of horizontal flame travel.

Low installation costs are possible because ELECTRUNITE is ductile, truly concentric, with exacting uniform wall thickness. It goes in fast, goes in easy. Quality is built-in. ELECTRUNITE is made of highest quality flat-rolled open-hearth steel produced in Republic's own mills to Republic's rigid specifications . . . quality controlled from ore to finished product.

Each length of Republic ELECTRUNITE Boiler Tubing is hydrostatically or electronically tested to conform with applicable ASTM specifications, the ASME Boiler and Pressure Vessel Code, as well as local, state, and boiler insurance requirements. Republic ELECTRUNITE is approved on an equal basis with tubes made by other processes up to 850°F. It is available for pressures over 2,000 psi, in various sizes and wall thicknesses.

To learn how Republic ELECTRUNITE can help you solve your boiler, condenser, heat exchanger tubing problems, call your Republic representative, or write direct.



FARROWTEST® is more than a quality control test. It detects and *rejects* defects of critical size. This quality protection is your proved production safeguard. Republic offers you the additional quality protection of exclusive FARROWTEST, as an alternate to other, less positive tests. It can be applied to your next order of Republic ELECTRUNITE Boiler Tubing.

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World's Widest Range of Standard Steels and Steel Products

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1441 REPUBLIC BUILDING • CLEVELAND 1, OHIO

Please send the following literature:

- ☐ Booklet—ELECTRUNITE Boiler Tubes
- ☐ Brochure—ELECTRUNITE Heat Exchanger Tubing
- ☐ Carbon Steel
- ☐ Stainless Steel
- ☐ FARROWTEST information
- ☐ Wall Chart on care and maintenance of boiler tubes

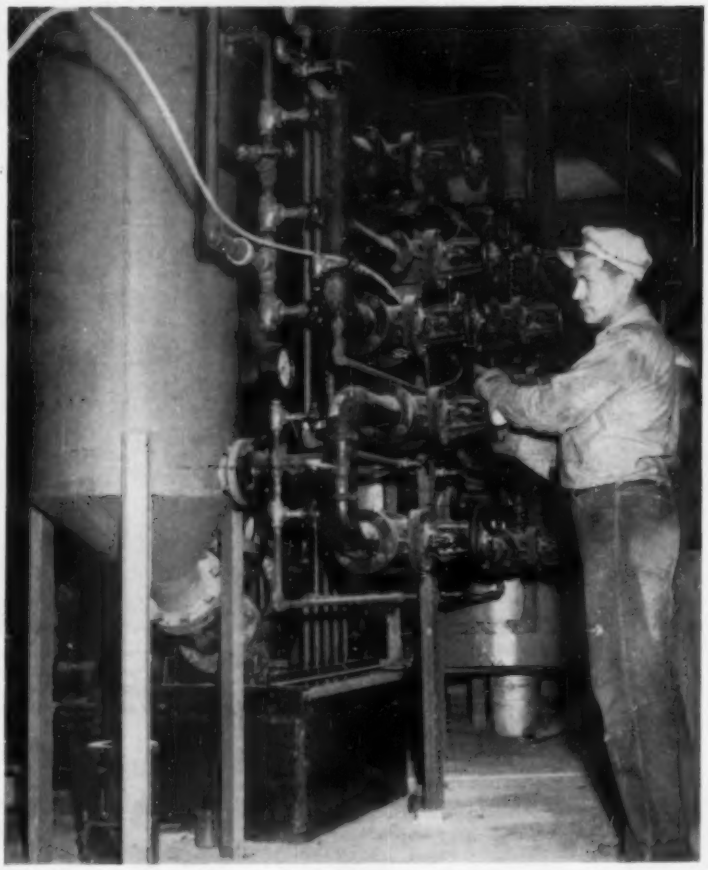
Name_____ Title_____

Company_____

Address_____

City_____ Zone____ State_____

Successful **ClaRite** Filters are adopted by **Supercritical Installations**



ULTRA-PURE FEEDWATER PROVES ECONOMICAL— WHERE IMPURITIES ARE MEASURED IN PARTS PER BILLION

Due to the excellent results obtained from ClaRite filters at the Ohio Power Company's Philo plant, Croll-Reynolds Engineering is now supplying other supercritical installations where high efficiency is required operating procedure.

Experience has proved that when ClaRite pre-filtration apparatus is installed with demineralization equipment, plant operations at high pressures and temperatures reach previously unattained results.

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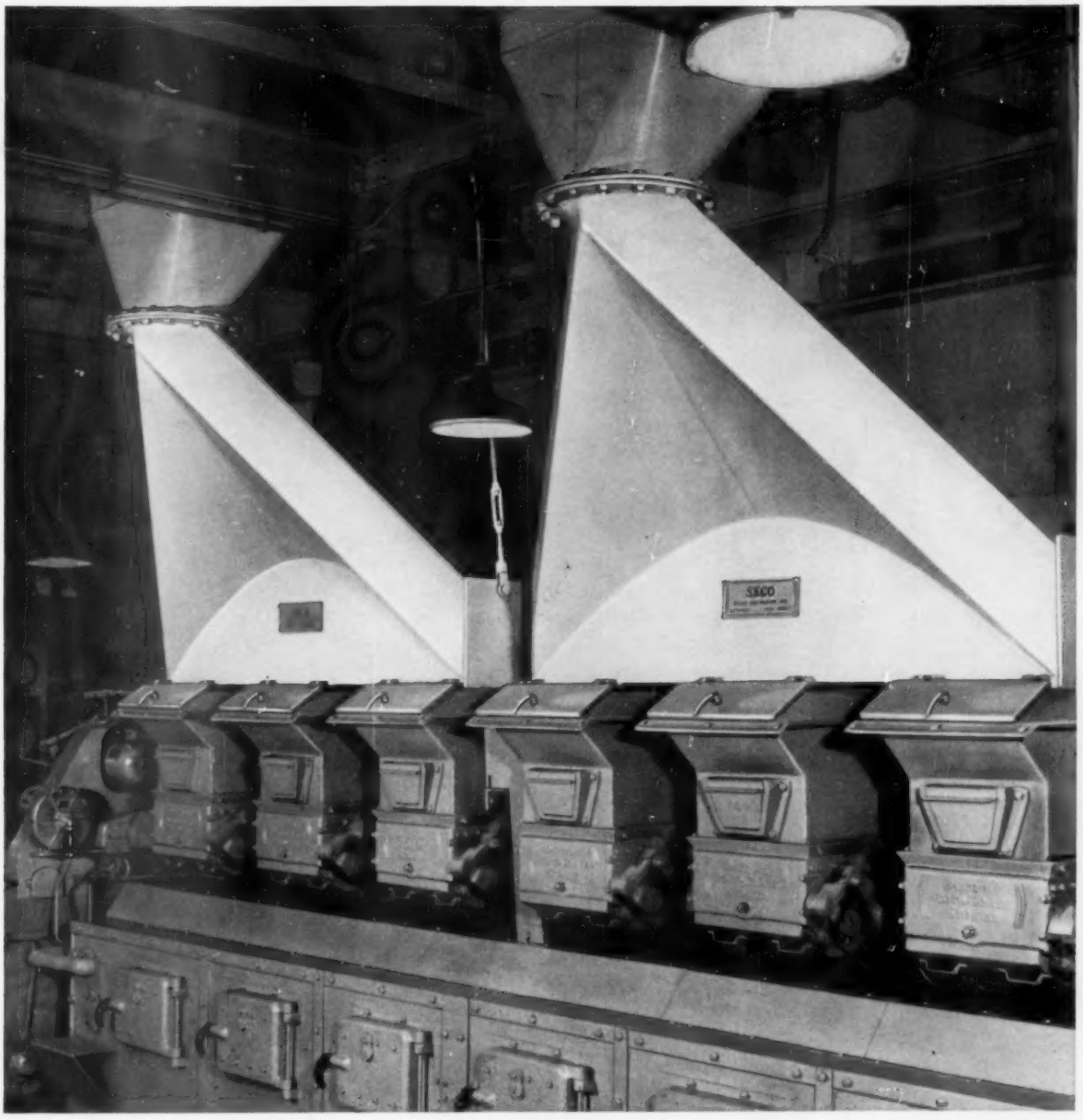

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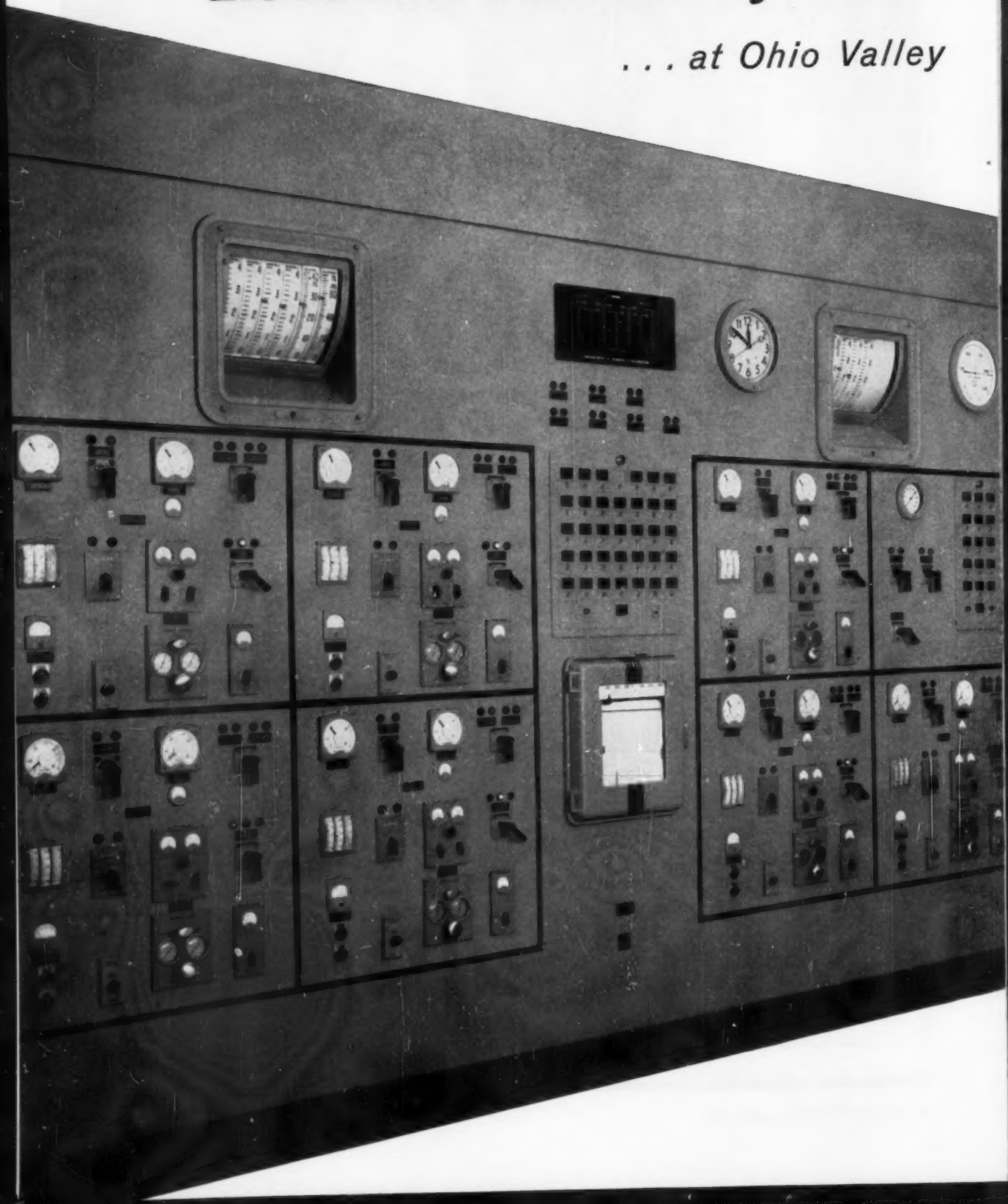

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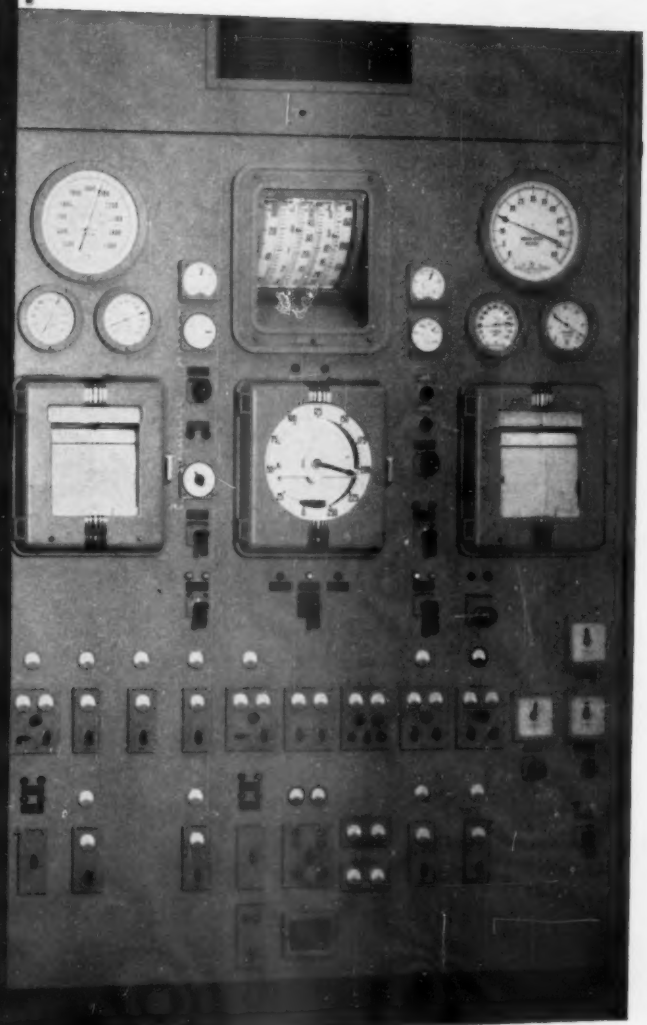

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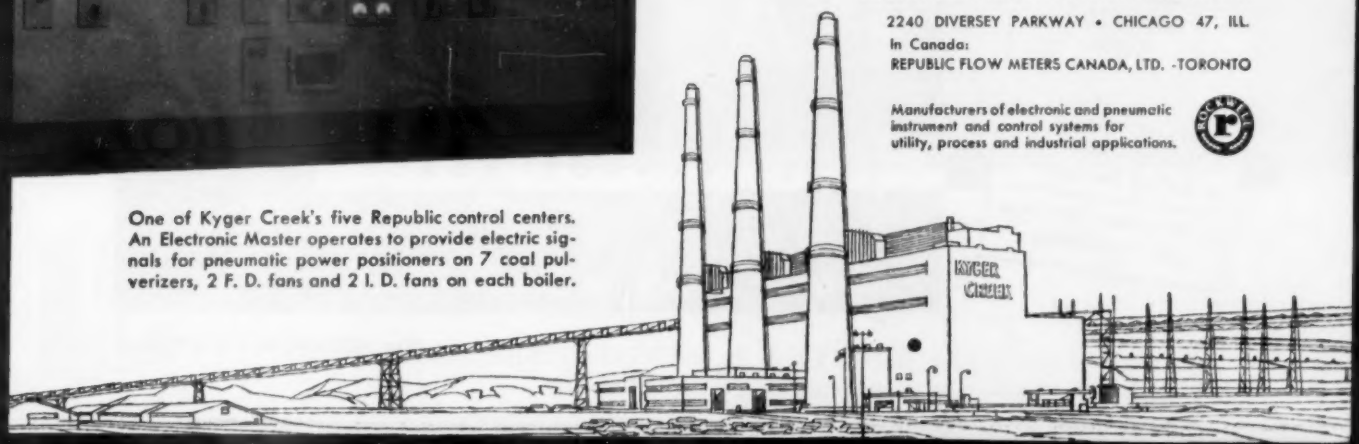

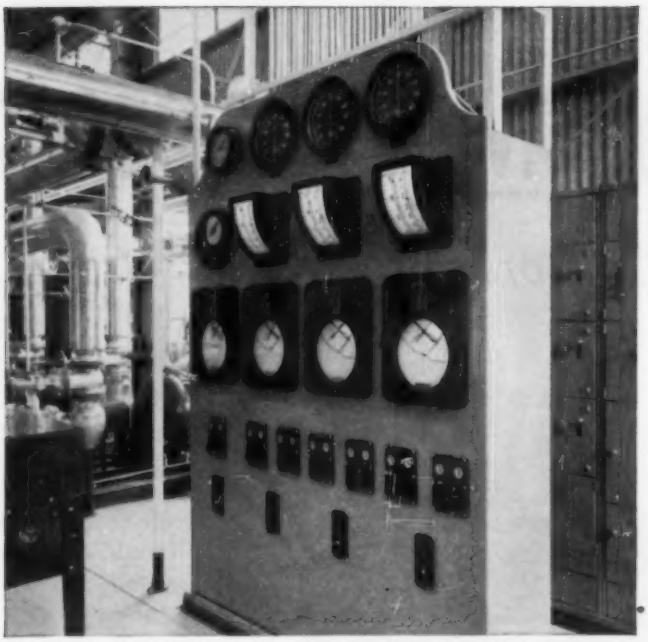

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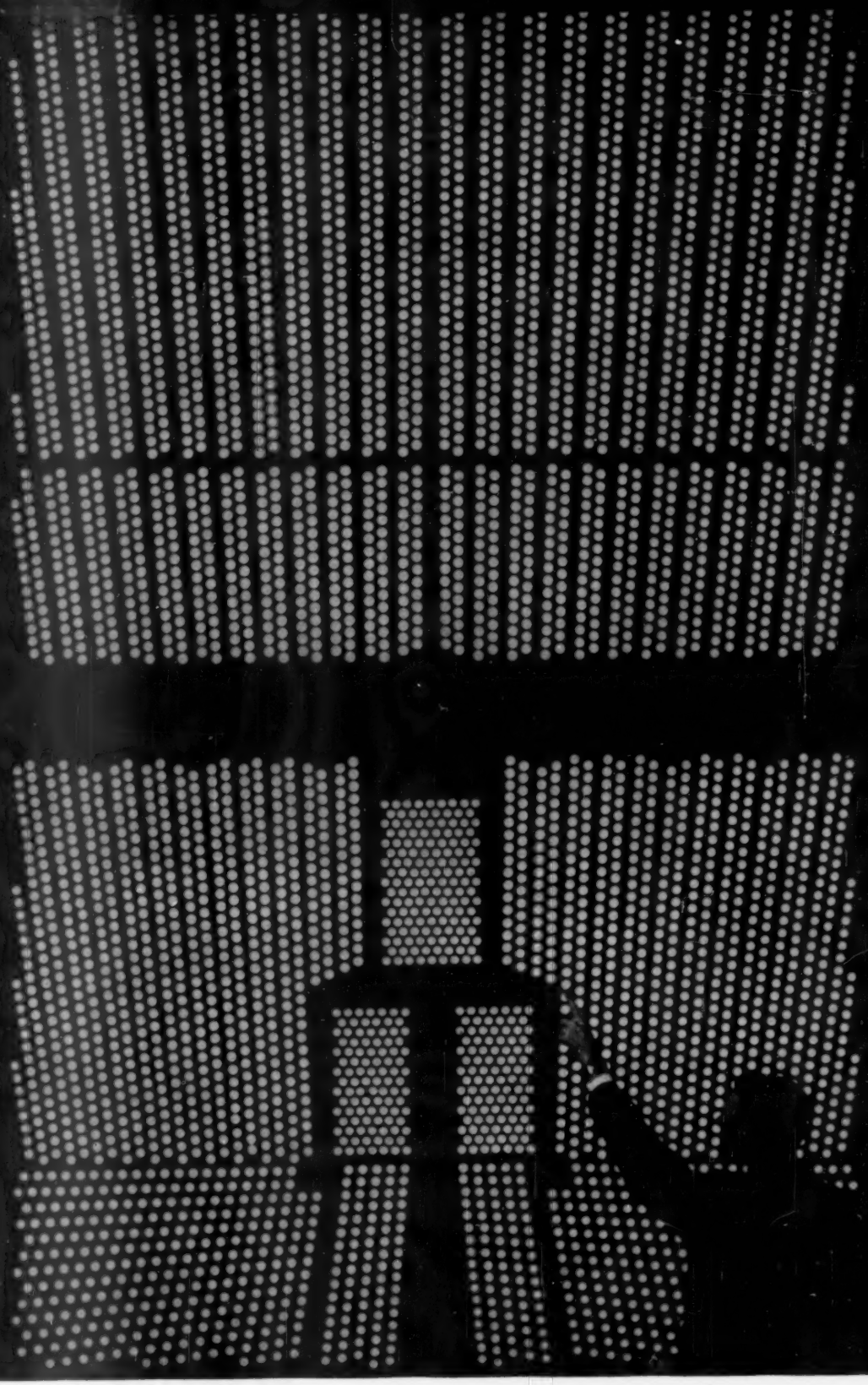

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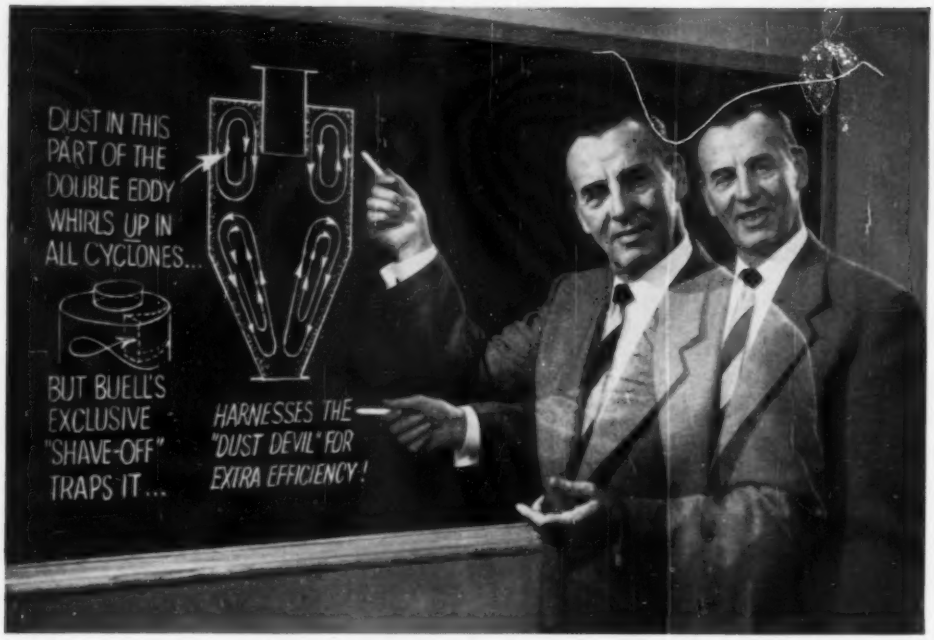
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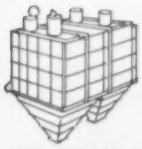

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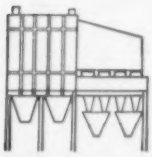

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Editorial

Pioneering and Power Plants

The responsibility of pioneering in power plants has become more and more a dual one shared by both the manufacturer of the primary equipment and the power plant owner employing that equipment. The impetus for design or cycle advances may originate with the plant owner's conviction that higher pressure, or higher temperature, for example, will produce economies of operation or with an individual manufacturer's certainty that a competitive edge will result from exploiting a new design or a cycle refinement. No matter what the origin of the idea its fruition depends upon cooperative participation of both principals—equipment supplier and equipment user. Today the power plant is so highly complex, and its primary elements so costly this cooperative participation needs the benefits of sustained programming and scheduling.

Over the past several years there have been many scattered appeals for a more active partnership in the programming of industry advances. Certainly the present recession with its effects on marketing practices of manufacturers is bound to affect any buyer-seller relationship. Our contemporary, *Electrical World*, in its March 10 editorial very clearly pointed out the dangers inherent to this relationship if another "white sale" reminiscent of a few years ago should develop.

But the partnership we feel necessary is not meant as one to protect the manufacturer from the vagaries of the marketplace but rather to permit him to intelligently plan his share of the industry's research and product development. For example, about two years ago, H. G. Ebdon, now the president of Combustion Engineering, Inc., brought the problem into sharp focus when he asked a predominantly central station audience the question, "What have you done to encourage more uniform ordering of boilers to avoid year-to-year extremes as in the past?" Mr. Edbon then sketched the effects of these extremes upon his own company's capital expenditures.

In good years or bad the manufacturer has to keep tooling up to stay in competitive position for the next wave of buying. Yet he cannot estimate with any assurance the load factor for his equipment for any span of time let alone determine a reasonable operating life for that same equipment. Obviously such a non-predictable production output reacts against research.

More recently at a seminar sponsored by his company, W. F. Crawford, president of the Edward Valves, Inc., took still another look at this problem. He referred to an earlier published interview of his whose title we have borrowed for this editorial. In brief Mr. Crawford had stated at this interview and reiterated at his recent seminar that poineering research and development were the lifeblood of our industry. Highly pertinent supporting facts were advanced for this conviction such as the one that of the total electric generating capacity in the U.S.A. 64 per cent of it has been installed within the past ten years! Further as an example of the effect of this consumer thirst for newer developments upon a manufacturer it was stated that within the Edward Valve family 60 per cent of their line is less than five years old.

Certainly, these facts bespeak a restless, active market. In its present state, however, that market imposes a heavy retooling burden upon its suppliers with largely speculative returns. Perhaps now is the time to establish through either the ABAI or one of the utility groups a long range development program with parts of this program assigned or allocated to segments of the supply industry so the benefits of programming can be applied to pioneering of power plants.

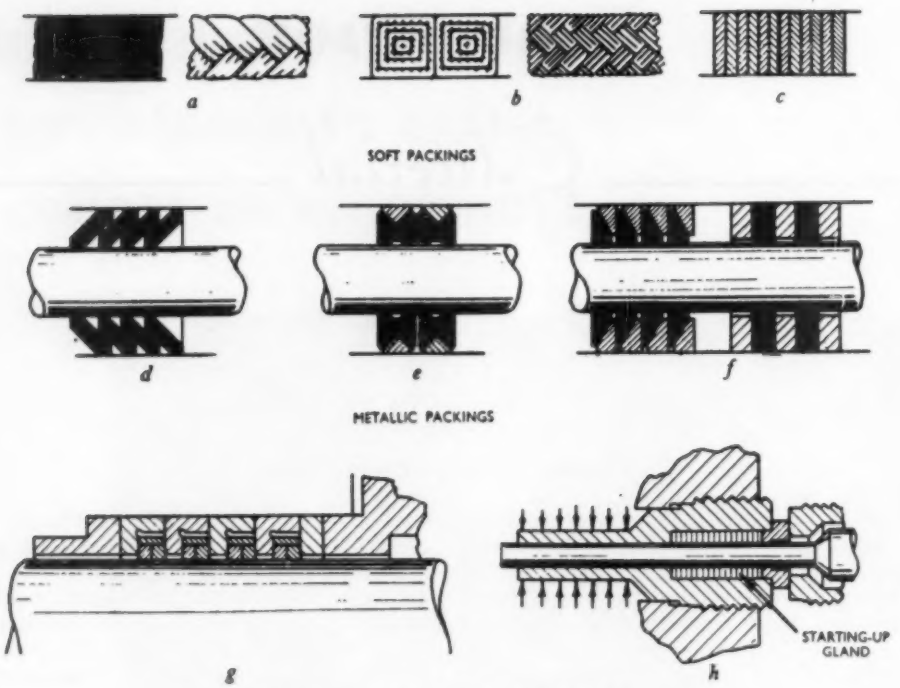


Fig. 1—Selection of packings in order of suitability for higher pressures

a—Plaited throughout.
b—Braided layer upon layer.
c—Laminated.
d—500–1000 lb/in.².
e—1000 lb/in.².
f—2–3000 lb/in.², 4–5000 lb/in.².
g—Multi-compartment gland with floating packing for pressures in the range 500–5000 lb/in.².
h—Extended-bush gland for pressures exceeding 10 000–50 000 lb/in.².

Packed Glands for High Pressures: an Analysis

THROUGHOUT the long history of engineering few problems have reappeared more frequently with each new development than that of effecting a satisfactory fluid seal between surfaces in relative motion. The pump, the steam engine, the hydraulic press, the screw propeller, the turbine, each in its turn has posed a new problem, clearly related to those which preceded it, but demanding a fresh outlook and a new solution.

In a previous paper by Beacham and Towler (1949),[1] a hydraulic seal was defined as one whose function it is to prevent the leakage of liquid between surfaces having relative motion. On this basis, many packed glands, especially those for high pressures, cannot qualify as seals, and a less restrictive definition is required. Such a definition must emphasize the maintenance of a pressure difference as the primary function of a high-pressure gland; leakage, although important, is secondary. It is convenient to borrow an electrical term and to regard a packed gland as a pressure-drop device of high impedance in which the leakage may be made as small as desired by making the impedance sufficiently high. Permissible leakage varies widely in different applications and it is unrealistic to demand zero leakage in all instances if this can be achieved only at the price of high friction, overheating, wear and excessive maintenance. In this paper the function of the gland is considered to be to control leakage but not necessarily to eliminate it, and various means of reducing leakage are considered. However, where some leakage is permitted, it is reasonable to expect that adequate provision will be made in the design of the gland to conduct away the leakage and return it to the system. Leakage is not important until the user becomes aware of it. The turbine and the oil-engine injection-pump are good examples from practice; by acceptance of slight leakage a difficult sealing problem is reduced to one of great simplicity. Here, fine clearances are used to control the leakage within acceptable limits, and the slight loss in performance is a small price to pay for the reliability and freedom from maintenance which is achieved. In some instances a high degree of fluid-tightness is an essential requirement, for example, in the oleo-pneumatic cylinders of the recuperator system of a gun or of an aircraft undercarriage, where loss of pressure and frequent recharging in the field would be unacceptable. Here, however, no formidable combination of factors arises and, by arranging that oil and not air is the fluid in contact with the gland, almost complete elimination of leakage is possible with relatively simple packings.

High pressure differences give rise to large forces which, suitably harnessed, can be the means of effecting the sealing action but which, alternatively, by causing unwanted distortion and excessive bearing loads, can be at the root of most of the troubles encountered. Thus the

[1] An alphabetical list of references is given at end of article.

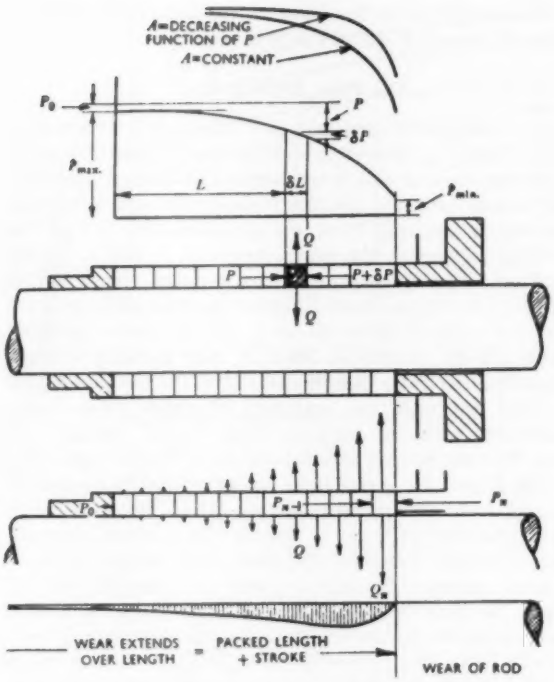


Fig. 2—Pressure and force distribution in the stuffing box

As the author so aptly states few engineering problems reappear more frequently with each new development than that of effecting a satisfactory fluid seal. Here is an attempt to develop an approximate theory as a basis for design, correlation of service data, or an indication for future investigation and research.

By PROFESSOR J. L. THOMSON†

Royal Military College of Science

# of Fundamentals*

solution of a gland problem is much more than the mere selection of the correct packing; it involves the rigidity and alignment of the whole machine, and at really high pressures the requirements of the gland will largely dictate the nature of the entire design from its earliest conception. Ideally, the rigidity of the packing used should be greater for higher pressures in order to resist the increased crushing forces, but the more rigid the packing the less tolerant is it to variations in alignment, and soft packings are frequently used at high pressures solely because of their adaptability to the deficiencies in the design of the machine. Heavy maintenance due to rod wear and frequent renewal of the packing is the price that must be paid. Fig. 1 shows a selection of packings arranged in order of suitability for higher pressures. The aim is to secure, by controlled deformation of the packing, just sufficient conformability at the working pressure. Reduced friction and wear together with improved thermal conductivity are obtained with metallic packings and the many designs available differ mainly in the means adopted to reconcile the requirement of adequate flexibility with sufficient rigidity to resist collapse. In the multi-compartment gland, Fig. 1*g*, the conflicting requirements are met by using floating packings isolated in separate compartments so that each element is relieved of all but a fraction of the total crushing force. Where really low friction is essential a clearance gland must be used. At high pressures, dilation of a plain bush causes the clearance to increase and some form of extended bush h must be used to counteract this. Careful selection of the dimensions and the elastic properties of the material of the bush is necessary to achieve adequate control of the clearance at the working pressure. This type of gland has negligible tolerance to misalignment. The original proposal was due to Bridgman (1931) and a modified design has been used by Morrison and others (1956).

The Clearance Gland

Before considering the packed gland it is worth while briefly to investigate the simpler clearance gland. If this is idealized as a rod in a plain bush with an annular clearance c, then, on the assumption that the flow in the clearance space is laminar, it can readily be shown that the leakage when the rod is stationary is given by

$$q_0 = \frac{\pi c^3 r}{6\eta} \times P/L \qquad (1)$$

where r is the mean radius of the annulus, η the viscosity of the fluid, P/L the pressure gradient.

Reciprocation of the rod in the bush will cause an increase or a decrease in the leakage according to the direction of translation, that is, with, or against, the flow.

* Presented before The Institution of Mechanical Engineers, Feb. 21, 1958 and published by Institution permission, mailing address, 1 Birdcage Walk, Westminster, London, SW1.

† Department of Civil and Mechanical Engineering, Shrivenham, Swindon, Wilts.

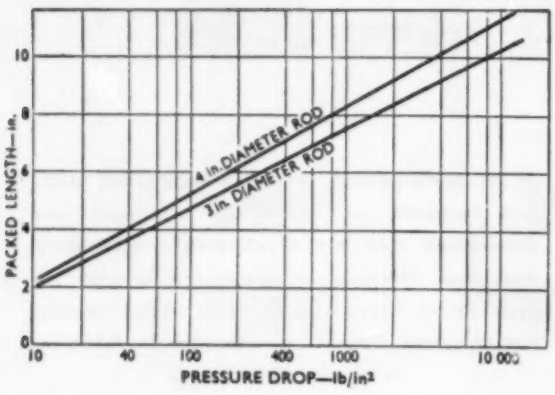


Fig. 3—Logarithmic plot of packed length-pressure drop relation

But the important point is that the leakage increases as the cube of the clearance, so that this factor dominates all others. It follows that the leakage will be greater if, for the same total difference $2c$ on diameter, the clearance is non-uniform and, again, it can be shown that when the rod is eccentric in the bush to the maximum extent possible, namely c, the leakage is

$$q = 2.5\, q_0 \qquad (2)$$

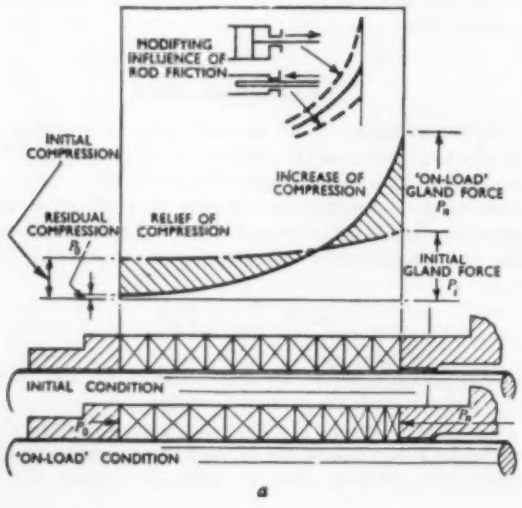
Consider a gland for a pressure of 1000 lb/in.² with a rod 4 in. diameter working with a mean radial clearance of 0.001 in. in a bush 10 in. long, so that the pressure gradient is 100 lb/in.²/in. For oil with a viscosity of 100 centipoises the leakage q_c would be of the order of 2 pints an hour. This might be acceptable, but for water or a fluid of similar viscosity the leakage would be a hundred times greater, that is, about 23 gal/hr, and this almost certainly would not be acceptable. If the clearance is reduced to 0.0001 in. the leakages become 0.002 pint (1 cu cm approx)/hr and 0.2 pint/hr respectively. This is the order of leakage which it is felt should be expected. But even the wider clearance of 0.001 in. is finer than the alignment of most commercial machines would permit and consequently some form of packed gland must, in general, be used if leakage is to be kept small.

The Plain Stuffing Box

It is commonly supposed that the sealing action of a packed gland is brought about by the initial tightening. This may be true at low pressures, but at high pressures the sealing action is automatic, being brought about by the forces resulting from the pressure drop across the packing elements, the effect being cumulative. Each packing element constitutes an annular piston across which there is a pressure difference and, in addition, each element supports those above it and the forces on them. Squeezed by this axial force P, the packing spreads laterally, exerting a sealing force Q on the rod and on the stuffing box wall, the intensity of which varies automatically with the pressure drop Fig. 2. While it is true that the sealing action is initiated by the tightening of the gland, the active force under operating conditions results from the pressure drop. Thus, for dimensions corresponding to those of the clearance gland example, the maximum value of the axial force would be about 5 tons, whereas the initial gland force would not normally be more than a fraction of this. This cumulative build-up of the forces results in the packing ring adjacent to the gland face, where the fluid pressure is least, being squeezed the hardest and, frequently, excessively so. This is clearly demonstrated by the way in which the wear, both of the rod and of the packing, is distributed. On this basis, and making two simplifying assumptions, it is possible to formulate a theory of the sealing action of a packed gland, to show that the distribution of pressure drop is substantially of exponential form and to derive an expression for the minimum packed length in terms of the pressure drop and packing and fluid to be sealed.

The sealing action arises from the lateral spread of the packing and, if Q is the lateral force exerted by the packing when squeezed by an axial force P[2] then we may

[2] For simplicity and to avoid confusion with the intensity of fluid pressure, p, unit areas are considered so that the pressure difference P gives rise to a force P. Also $dP/dL = -dp/dL$.



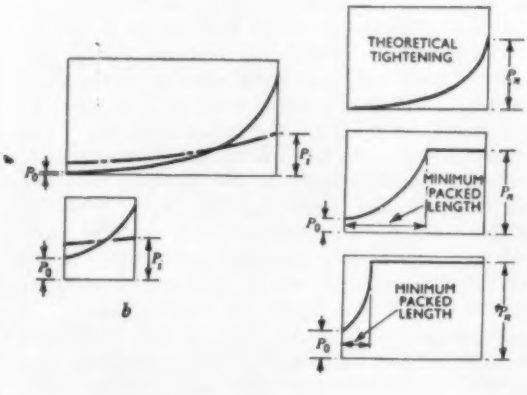


Fig. 4—Influence of initial tightening

a—Redistribution of compression in packing. *b*—Comparison of deep and shallow stuffing boxes. *c*—Development of pressure intensification with increase of P_0.

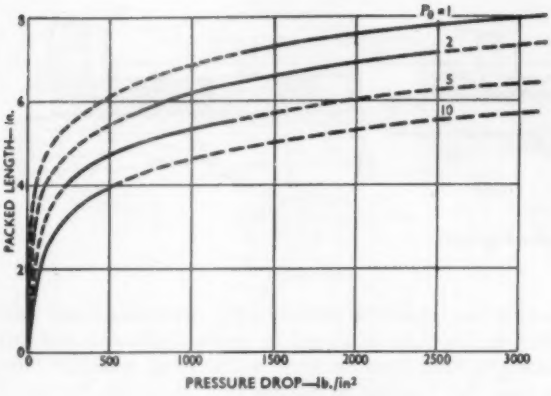


Fig. 5—Packed length-pressure drop relation

$L = 2.3 \log P/P_0$.

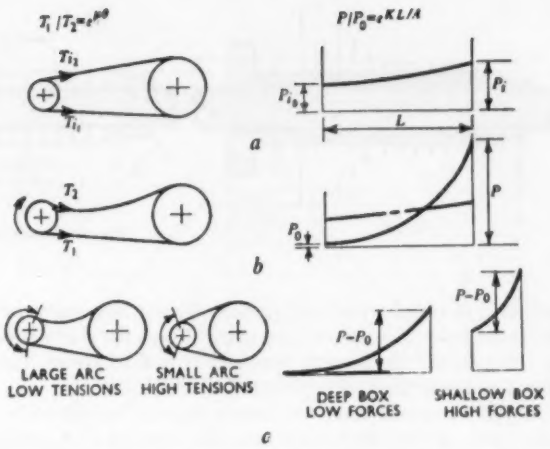


Fig. 6—Comparison of belt drive and packed gland

a—Initial condition.
b—"On-load" condition.
c—Alternatives for the same duty.

write $Q = f(P)$ and to a first approximation we may assume that

$$Q = KP \tag{3}$$

where K is a constant depending upon the material and form of the packing. The value of K will be smaller the more rigid the packing and will tend to unity for soft or plastic packings. Thus the packings of Fig. 1*a* to *f* are arranged roughly in order of decreasing values of K. Strictly, K is not independent of P and the value for all packings will tend to unity at high stress, as all materials behave plastically if the stress is high enough. Squeezed against the rod surface with a force Q the element of packing will be able to establish a limiting pressure gradient dP/dL, Fig. 2, and the second assumption is that a relationship will exist such as

$$dP/dL = Q/A \tag{4}$$

where A is a constant with a value depending upon the conformability of the packing, the smoothness of the surface of the rod and physical properties of the fluid, such as viscosity and surface tension.

Combining (3) and (4)

$$\frac{dP}{dL} = \frac{KP}{A}$$

hence

$$\int dL = A/K \int \frac{dP}{P} + \text{const}$$

that is,

$$L = A/K \ln P + C \tag{5}$$

or pressure drop[3]

$$P = Be^{KL/A} \tag{6}$$

It will readily be seen that the constant C (and hence B) is dependent upon the degree of initial tightening of the gland, provided that this has been sufficient to produce residual compression at the face of the neck bush under working conditions. The effect of such residual compression is to reduce the packed length required to produce a given pressure drop. At the present time there are virtually no data giving discreet values of the constants A and K for particular packings working with particular fluids. But plotting the relationship expressed by equation (5) yields straight lines if a logarithmic scale is used for P, and such information as is available can best be presented in this way, Fig. 3. The influence of the many variables previously mentioned makes comparison of results from ordinary industrial service difficult, and progress would be aided by the existence of a reference line based on results obtained under standard conditions. These would include limits for leakage and for misalignment or eccentricity of the rod in the stuffing box. But in many instances it is unnecessary to evaluate the constants. Results at one pressure will exist and an

[3] *Strictly, pressure drop* $= P - P_0$.

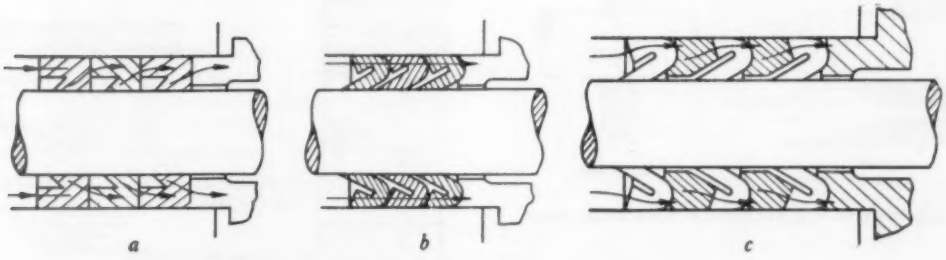


Fig. 7—Transmission of axial compressive force through packing

a—"Automatic" hydraulic packing.
b—S.E.A. rings.
c—S.E.A. rings with formed spacers.

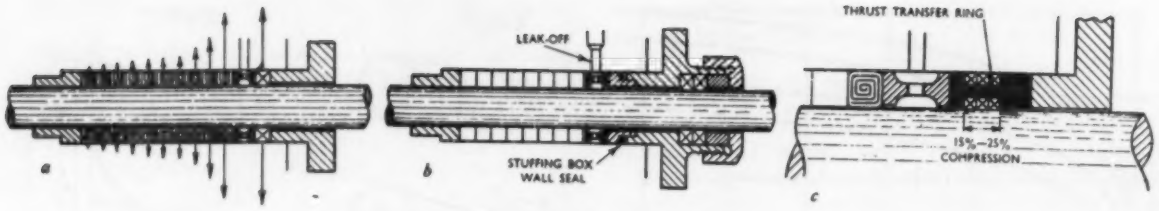


Fig. 8—Glands with leak-off lanterns

estimate is required of the packed length necessary for the same fluid at another pressure. Thus, for a pressure of 1000 lb/in.^{2} the length required is not 10 times, but only $1^1/_2$ times that for a pressure of 100 lb/in.^{2}.

It will be shown later that there is good reason to believe that, at very high pressures, the value of A/K tends to unity, and for soft packings this will be approximately true at pressures of 4–500 lb/in.^{2} and greater. In these circumstances equation (5) reduces to

$$L = 2.3 \log P + C \qquad (7)$$

Putting $C = 0$, this simple relationship gives quite good correlation with normal practice but, rather surprisingly, it gives greater lengths than are actually found to be necessary at low pressures (when A/K would be greater than unity), especially for rods of small diameter. There seems little doubt that this is because the significance of the constant C becomes greater in these instances. From equation (5) the value of the constant is clearly

$$C = -A/K \ln P_0$$

where P_0 is the residual compression of the packing at the face of the neck bush, that is, at the point where the maximum fluid pressure p_{max} exists. Compression at this point is produced initially by tightening of the gland, but a redistribution occurs under working conditions and, a result of the increased compression of the packing at the gland end, relief of compression occurs at the neck bush end, Fig. 4. It will be apparent that the probability of the existence of high residual compression at the neck bush will be greater, (1) for shallow stuffing boxes and (2) for small diameter rods. Substitution of the above value of C in equation (5) yields

$$L = A/K \ (\ln P/P_0) \qquad (8)$$

In this form it will be seen that the effect of a small variation in the value of P_0 can be quite significant, and any standard conditions for tests would most certainly have to take this variable into account. However, the effect on the packed length required is less critical than might appear at first sight, especially at high pressures, and in Fig. 5 curves given by equation (8) have been plotted for various values of P_0. The curves are shown as full lines in the regions where the appropriate value of P_0 might be considered to apply. With some sets of proprietary packings, usually metallic packings, springs are included to maintain the residual compression at the neck bush end and, to some extent also, to avoid overcompression of the packing.

Frictional forces will also have a modifying influence on the build-up of the axial force P. To allow for rod friction it is necessary to introduce a factor $(1 \pm K\mu)$, the sign used depending upon the direction of rod movement as indicated at the top of Fig. 4.

Rearrangement of equation (8) gives

$$P/P_0 = e^{KL/A} \qquad (9)$$

In this form the similarity of the basic relationship for the packed gland with the more familiar relationship $T_1/T_2 = e^{\mu\theta}$ for the belt drive will be apparent and, as Fig. 6 shows, the physical correspondence, which is equally close, helps toward an understanding of the functioning of the packed gland.

Figs. 2 and 4, and the foregoing argument, make it clear that packing rings at opposite ends of the stuffing box are subjected to quite different conditions; a total compressive force per ring of, perhaps, 50 lb in one instance and 5 tons or more in the other. Further, the fraction of the total pressure drop due to each ring varies roughly in the same proportion, from about 0.2 per cent

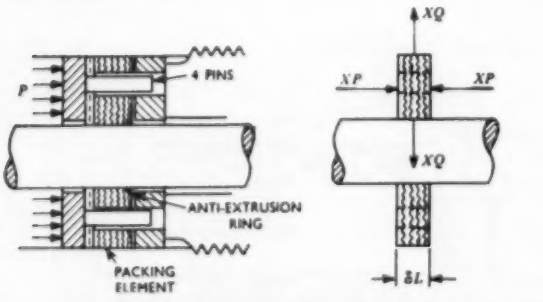


Fig. 9—Unsupported-area-type gland used in liquid spring, above

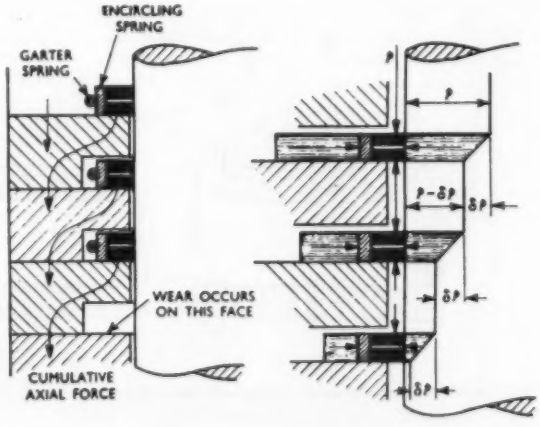


Fig. 10—Multi-compartment gland-forces and pressure distribution

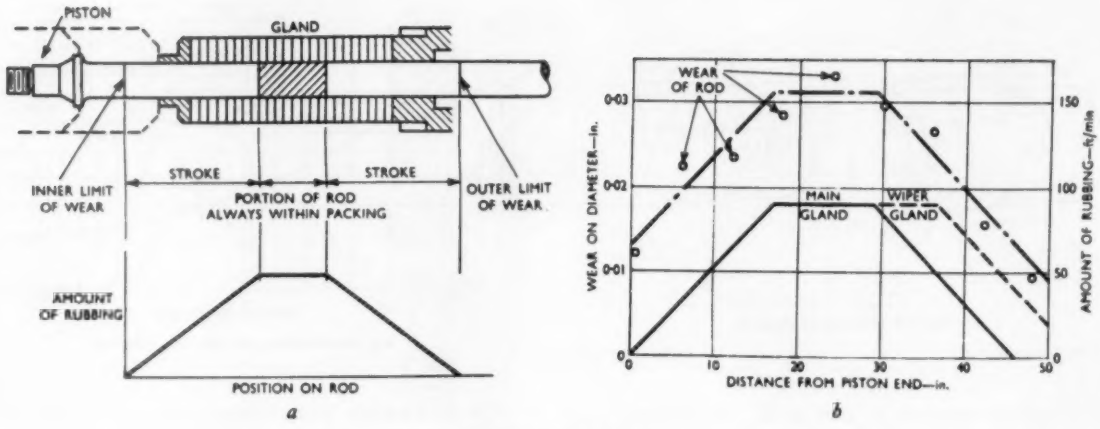


Fig. 11—Wear of rod

Multi-compartment gland, 4000 lb/in.² gas circulator.

for the first ring to about 50 per cent for the last ring adjacent to the gland face. This high pressure drop per ring is achieved because the force Q, with which the packing is squeezed against the rod, is correspondingly high and at high pressures this force tends to be excessive, leading to scoring of the rod and rapid wear. Greatly extended life might be expected if it were possible to equalize the work done by the various packing rings. Nothing can prevent the cumulative build-up of compression from ring to ring, but the use of progressively harder packings, that is, with lower values of K, would prevent a corresponding build-up in the value of Q. Further, such packings would compress less and, in consequence, would cause less relief of compression at the opposite end of the box. A rational packing set for a high-pressure stuffing box would consist, therefore, of rings of graded K values, the packing varying from hard at the gland end to soft and springy at the neck bush. Although not designated by their K values, packings with an appropriate range of properties already exist and some of these are shown in Fig. 1. Intermediate in properties between *b* and *c* come the range of foil-wrapped packings which are manufactured in grades from soft to hard, depending upon the nature of the core and the number of layers of foil. Minimum lateral spread occurs with laminated packings, Fig. 1*c* which may be quite hard; the asbestos sleeves of sleeve-packed cocks and valves being, perhaps, the best-known examples. An alternative to making the whole packing ring harder, is to design a portion of it specifically to carry the axial load while leaving the sealing portion flexible. The block portion of the packing shown in Fig. 7*a* in which transmits the axial force while each lip is left free to function independently. For the same reason, wire-mesh reinforcement is usually incorporated in the heel of the packing shown in *b*, and performance of this type can be still further improved by the use of formed spacers as shown in *c*, which transmit the axial force from ring to ring with less interference with the lip action.

Two factors largely prevent the extensive use of harder packings in practice: (1) variations in alignment of the rod in the stuffing box, and (2) inadequate depth of the stuffing box. Angular distortion under load of the type shown in Fig. 16 results in maximum movement at the very point at which it would be desired to use the harder packings. Equations (5) and (8) show that the packed length is directly proportional to A/K and the value of this ratio is high for hard packings, so that longer stuffing boxes are required for the same pressure drop. In consequence, machines designed for use with soft packing

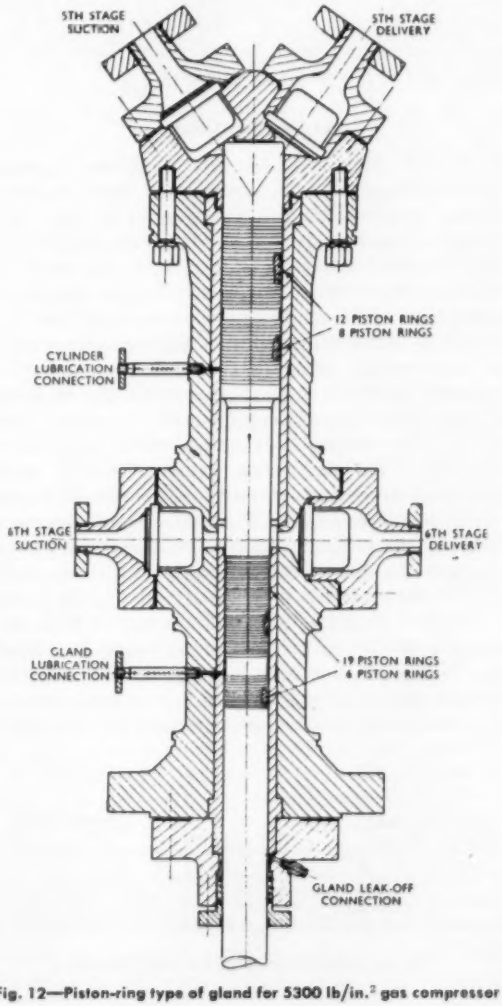


Fig. 12—Piston-ring type of gland for 5300 lb/in.² gas compressor

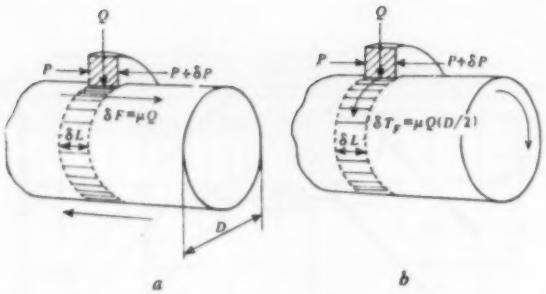


Fig. 13—Friction of element

a—A reciprocating rod. *b*—Rotating shaft

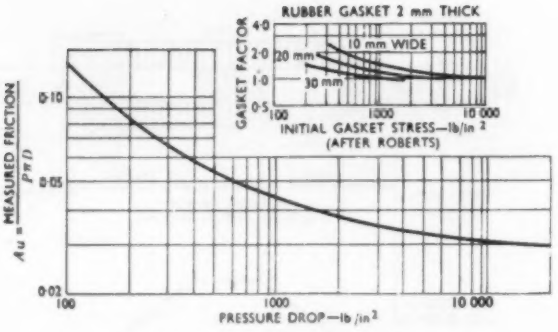


Fig. 14—Friction-pressure drop relation

are not at all receptive to new ideas. A common practice which is completely at variance with the theoretical ideal is the use of a turn of soft packing in the main stuffing box to seal a leak-off lantern when this is fitted Fig. 8. The full compressive load, of several tons, is transmitted through this packing from the lantern to the gland, although the fluid pressure is negligible and much wear and overheating result. Where leak-off lanterns are fitted, and an extension of their use is to be encouraged, a separate low-pressure stuffing box should be used as in *b*. In favorable circumstances the simpler arrangement as in *c* is satisfactory. Here the two portions of a thrust-transfer ring butt together and limit the compression of the low-pressure packing.

Pressure Intensification

So far very little has been said about the constant A in the equations. This expresses the ratio of the normal force Q to the resulting pressure gradient. It has for long been an accepted principle in connection with the design of bolted joints that the bolt force must be such as to ensure that the gasket stress is higher than the fluid pressure and the term "gasket factor" is in use for the ratio of the gasket stress to the fluid pressure at which a leak commences. Roberts (1950) has suggested that the gasket factor is a characteristic of the material of the gasket, the values being, for normal pressures, of the order of 2 for rubber, 5 for leather, 50 for lead and several hundred for copper, the value decreasing with increase of pressure and tending to unity for all materials at very high pressures. Thus the gasket factor is a measure of the pressure intensification necessary to effect a seal and the value decreases with increasing conformability of the material. This is also true of the constant A. While it would be unwise at this stage to carry the analogy with the gasket too far, it would be surprising if the fundamental sealing mechanism could not be related, the gasket having to conform to a fixed surface and the packing to a surface that is continually changing.

The well-known Bridgman closure utilizes the unsupported-area principle in order to obtain the necessary pressure intensification, and the same principle has been applied in the design of the highly successful high-pressure gland for the liquid-spring type of oleo leg described by Bingham (1955). The characteristic feature of this gland, Fig. 9 is that it has only one ring of packing and, at first sight, it might appear that it was fundamentally different from the plain packed stuffing box. But its sealing action is identical. Either the entire pressure drop P may be caused by a single ring, in which case it is necessary to increase the axial and other forces in the ratio given by the intensification factor X, or the axial force may be limited to P, in which case the maximum pressure drop per ring (at the last ring) is P/X, so that more than one ring is necessary, as in the normal gland. With the intensified pressures, high surface finish and hardness of the rod and special means to prevent extrusion of the packing are essential. Applied to the liquid-spring gland, equation (4) becomes

$$\delta P/\delta L = XQ/A$$

But in this instance δP, the pressure drop per ring, is equal to P

so that $$P/\delta L = XQ/A$$

and as before $$Q = KP$$

Hence $$X = A/K\delta L \qquad (10)$$

For this gland the packing ring is of rubberized fabric, the rod is hard chromium plated, pressures are high and the fluid is oil, so that only relatively low values of the intensification factor are necessary to ensure good performance. The gland is admirably adapted to its function and constitutes an example of sound design.

The Multi-Compartment Gland

With liquids, the fluid itself provides lubrication and

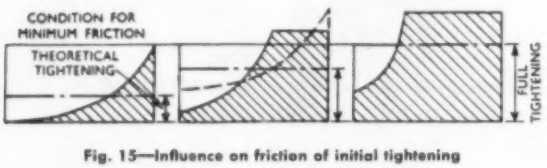


Fig. 15—Influence on friction of initial tightening

Friction is proportional to the shaded areas.

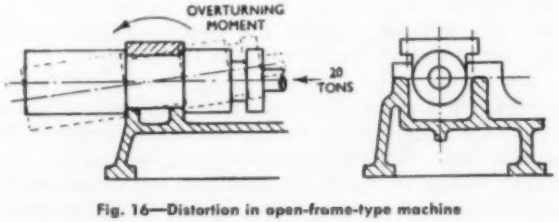


Fig. 16—Distortion in open-frame-type machine

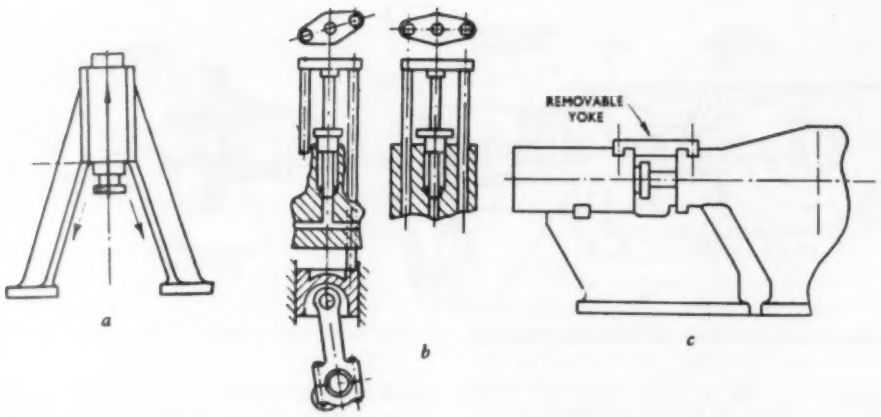


Fig. 17—Machine types *a*—Inherent symmetry of A-frame design. *b*—Aldrich-type pump. *c*—Semi-open frame design.

usually some measure of cooling, but with gases, especially where there is compression, the fluid heats the gland. Rod speeds are higher because of higher permissible gas velocities and it is necessary, therefore, to reduce to the minimum frictional generation of heat and to obtain good thermal conductivity. In large sizes particularly, this requirement is met almost exclusively by the multicompartment gland using metallic packing of the floating type. The essential feature of this type of gland is that the packing elements, by virtue of their isolation in separate compartments, are relieved of the cumulative build-up of the axial force, which is transmitted through the walls of the compartments. Each element is subjected merely to an axial force due to the pressure drop across itself, and if this is not excessive the element will be free to float with the rod. This imparts a tolerance to changes in alignment which it is otherwise difficult to obtain with metallic packing, but the floating feature brings with it its own difficulties, namely the sealing of alternative leakage paths. Accordingly there is considerable variety in the design of packing elements, the aim being to reconcile a number of requirements, such as, adequate masking of the joints, compensation for wear, robustness and ease of manufacture. Some of these designs are discussed in greater detail in the section dealing with leakage.

The sealing mechanism of the packing elements is similar to that of the piston ring, in that the sealing force comes mainly from the pressure acting on the top and back of the ring. Pressure intensification arises, in the case of the piston ring, from the initial "spring" of the ring into the bore and, in the case of the packing elements, from an encircling spring which may be of the "garter" type or of the piston-ring type, when it usually serves also to mask the radial joints in the packing. Increase of the contracting force increases the limiting pressure drop due to each element. It will be obvious from Fig. 10 that the effective pressure causing contraction of the elements onto the rod is the pressure drop across the element and not the local intensity of fluid pressure. The pressure drop per element is comparable with that for piston rings but, as with the latter, practice varies widely. Thus a figure of 100 lb/in.^{2} might be taken as an average for a total pressure drop of 1500 lb/in.^{2} and 200 lb/in.^{2} for a pressure drop of 4000 lb/in.^{2}, the tendency being to reduce the pressure drop per element for low pressures and to increase it for high pressures, so that the number of elements and compartments varies from about 5 to 20 for the entire range of pressures covered. Under constant or near-constant pressure conditions the loading on the elements is reasonably uniform throughout the length of the packing. Under pulsating pressure conditions, which are more usual in practice, the elements nearest to the neck bush bear the brunt of this pulsation, which is progressively damped out as the gland end is approached. The gland of the double-acting diesel engine is subjected to an unusually severe pressure variation and results obtained by Jones (1933) from indicator diagrams for a 6-compartment gland gave pressure drops of the order of 280, 120, 55, 18, 55 and 2 lb/in.^{2} respectively. Somewhat similar results have been obtained by Eichelberg (1940) for the pressure drop across a series of piston rings; values for a maximum cylinder pressure of 300 lb/in.^{2} being of the order of 140, 60, 30, 25, 15 and 22 lb/in.^{2} respectively. But these represent transient conditions and the values over the greater part of the cycle are substantially less than the higher figures suggest. This pressure variation as given by the ratio of the peak cycle pressure to the mean pressure and its effect on the pressure gradient through the packing is a factor to be considered when determining the number of compartments required for a given pressure drop. With high-pressure gas compressors, the final gas temperature usually limits the compression ratio to about 2.5 which contrasts with a pressure ratio of about 35 for the diesel engine, so that although the total pressure may be greater, the range of variation within a cycle is relatively much less.

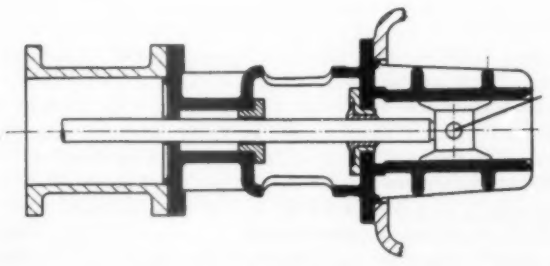


Fig. 18—Unit construction of crosshead guide and stuffing box

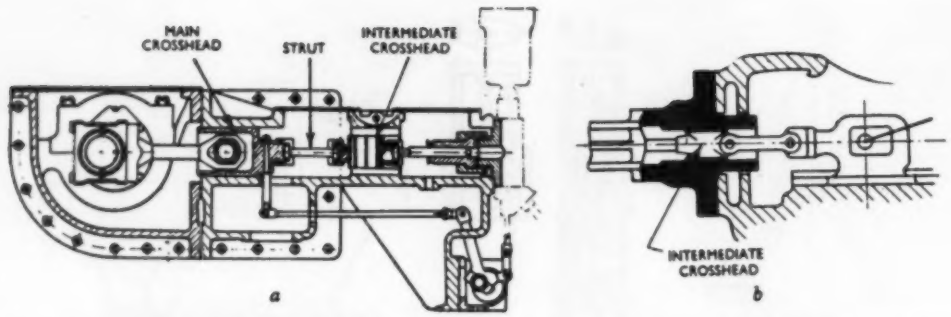


Fig. 19—Use of intermediate crosshead to maintain alignment *a*—Marine Diesel injection pump. *b*—High-pressure gas compressor.

Indirect evidence that the pressure drop in a multi-compartment gland is uniform under constant, or near-constant, pressure conditions is provided by the wear of the rod which, as shown in Fig. 11, bears a direct relationship to the amount of rubbing. This should be contrasted with the rod wear in an open-type stuffing box as shown in Fig. 2.

The pressure drop across the element also determines the force with which the element is pressed against the supporting surface and in this way governs the frictional resistance to the floating action of the packing. Wear at this surface is one of the factors limiting the life of the packing and, therefore, for a given life, the pressure drop and the amount of float are interrelated. For proper sealing action it is important that this supporting surface is ground truly flat and normal to the axis of the rod. The boxes forming the compartments should be fitted as indicated so that this surface is an external one. Although this may make installation of the packing more difficult, reconditioning of the boxes by regrinding of the supporting surface is greatly facilitated. With a high pressure drop per compartment, deflection of the supporting surface, that is, dishing of the compartment roof, may occur, and thoroughly robust construction is essential to cover the possibility that any one element may be required temporarily to sustain a pressure drop of two or three times its rated value.

It is well-established practice for the final stage of a high-pressure gas compressor to consist of a ram or plunger fitted with up to 20 piston rings. In recent years this practice has been extended, and piston rings mounted in the rod, as shown in Fig. 12, are now widely used as an alternative to the multi-compartment gland. This practice shows to greatest advantage at very high pressures, where the length of gland is large in relation to the piston stroke. Otherwise, the piston-ring gland necessitates a considerable increase in the overall length or height of the machine, as the minimum length of the gland liner is equal to the distance over the extreme rings plus the travel of the rings. The gland requires a highly finished and, therefore, expensive liner in which the rings work, but to offset this, the rings themselves are considerably cheaper than the sets of carefully hand-fitted packings, as there is a specialist industry producing piston rings on a quantity basis to extremely high standards of accuracy. Apart from the question of capital and replacement costs, there are important differences in characteristics. The multi-compartment gland is a commendably tight gland, in the sense that gland leaks are negligible for a large proportion of the life of the packing, whereas with the piston-ring gland there is a measurable leak throughout its life. On the other hand, the latter gland is markedly superior from the standpoint of economy of lubricating oil, and the corresponding reduction in oil contamination of the gas stream may frequently be of even greater importance.

Both types of gland rely on an adequate supply of oil to effect a gas seal but, whereas the moving piston rings distribute a restricted supply of oil fairly uniformly over themselves and over the surface of the liner, the fixed packing elements tend to scrape the oil from the rod, and the packings must be flooded with oil to prevent loss of the gas seal.

The piston rings are usually of the 2-turn Clupet type and a pressure drop per ring of up to 200 lb/in.^{2} is normal practice at 5000 lb/in.^{2} total pressure.

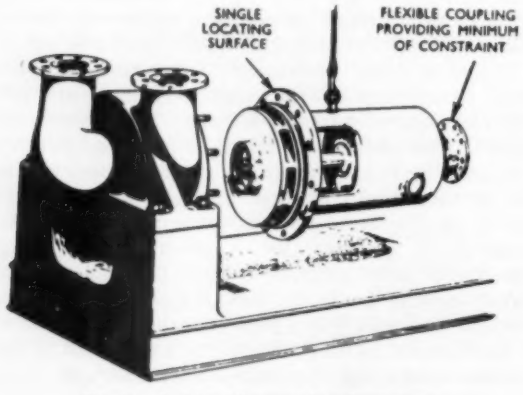


Fig. 20—Positive alignment in refinery pump

Friction

That friction is variable, even indeterminate, is generally regarded as being characteristic of packed glands; it being possible for the friction under apparently similar conditions to vary by a factor of 3 or more. Unfortunately there is no generally accepted method of expressing or comparing the frictional loss. In favorable circumstances the frictional resistance can be measured without great difficulty, but how, for instance, can the value of the coefficient of friction be determined from this measurement? Most frequently the friction is expressed as a pressure loss, but rods of different diameter cannot be compared on this basis. A more rational method is to relate the friction to the force normal to the rod surface, that is $F \propto \mu Q$

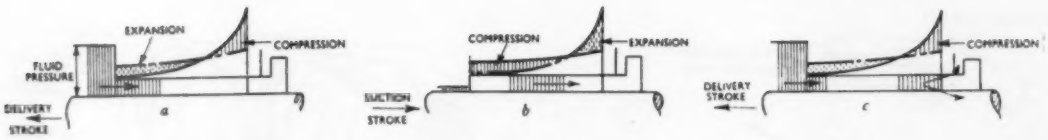


Fig. 21—The leakage cycle of sponge action *a*—Absorption. *b*—Transfer. *c*—Exudation.

Considering a narrow zone of width δL, Fig. 13

$$\delta F = \mu Q \pi D \delta L$$

But

$$Q = A\delta P/\delta L$$

Therefore

$$\delta F = \mu \pi D A \delta P$$

$$F = \mu \pi D A P + \text{const} \qquad (11)$$

As Q depends upon the constant K, it might be expected that K should appear in the expression for friction. But a lower value of K for the packing means that a greater packed length is required and hence that the frictional resistance is more widely distributed but is unchanged in magnitude. This is not to imply that the value of K is unimportant, because trouble arises from the concentration of friction, wear and heating, especially with rotating shafts. The value of the constant A would, in general, be higher for a metallic than for a fabric packing, but the effect on friction is offset by the lower value of μ for the former, and it is the product $A\mu$ that is significant. Again, the constant A influences the packed length required for a given pressure drop, and from the above relationship it will be apparent that A has the dimensions of length.

For any one rod-packing-fluid combination, equation (11) can be written

$$F = \text{constant} \times D \times P$$

and this is in agreement with general experience at high pressures. If the pressure range is sufficiently wide to embrace both high and medium pressures, however, it is found that F is not strictly proportional to P, but that there is an apparent decrease in the value of μ with increasing pressure. While this possibility cannot be ruled out, it is more probable that the observed decrease is in the value of the product $A\mu$. Earlier, from the analogy with the gasket joint, the constant A was compared to Roberts's gasket factor "m," the value of which tended to unity at very high pressures. If, from observed values of F at different pressures, $A\mu$ is computed, curves as shown in Fig. 14 are obtained and, from the similarity with curves for the gasket factor, it is difficult to escape the conclusion that A rather than μ is chiefly responsible for the variation. Values for gases are appreciably higher than for liquids, such as oil, and this is in line with the higher value of A required, but the poorer lubrication and its influence on μ may also be a factor, as with liquids the values of μ are definitely low, due no doubt to the forced lubrication provided by the fluid under pressure.

Recent correspondence in the technical press suggests that equation (11) might be objected to on the grounds that it indicates that the friction should be determinable with some degree of precision, whereas this is not in line with common experience. However, it is implicit in this equation that the pressure distribution in the stuffing box is as shown in Fig. 2 and this value of F is the minimum value. Excessive initial tightening of the gland will produce a different pressure distribution and higher friction. This is shown in Fig. 15 and some calculated values are given in Table 1, the basis for the calculations being a 4-in. diameter rod with $^3/_4$ in. square packing.

It can be shown that when the initial tightening force exceeds that corresponding to the pressure distribution in Fig. 2, the friction increases directly as the tightening force and is independent of the fluid pressure. This has frequently been observed and has been confirmed in experiments reported by Denny (1952).

The limiting case is considered to be when the initial tightening force is equal to the full pressure force, but this is, of course, a purely arbitrary limit. When the packing is uniformly compressed with this force, the friction is increased in the ratio

$$F/F_{min} = \ln P \qquad (12)$$

At low pressures the gland forces required are so small that it is difficult to avoid over-tightening but, as Table 1 shows, these forces attain such large values at high pressures that in practice the influence of initial tightening rapidly diminishes and becomes insignificant at very high pressures.

Table 1. Friction in Packed Stuffing Boxes

Pressure drop, lb/in²	20	50	100	500	1000	5000
Initial tightening (theoretical) Pressure	6·68	12·8	21·8	90·5	145	588
Initial tightening (theoretical) Total force	74·6 lb	143 lb	0·11 ton	0·45 ton	0·72 ton	2·9 ton
Full pressure force	224 lb	560 lb	0·5 ton	2·5 ton	5·0 ton	25·0 ton
$A\mu$ (from Fig. 14)	0·30	0·20	0·14	0·053	·044	0·032
Friction force, lb. Minimum	75	126	176	334	554	2010
Friction force, lb. Full tightening	225	493	810	Considered not applicable		
Ratio F/F_{min}	3·0	3·9	4·6	—	—	—

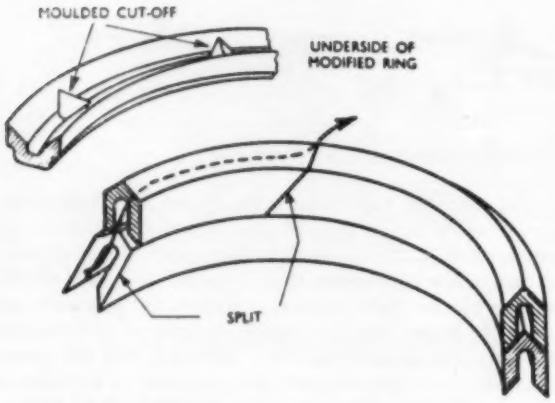


Fig. 22—Circumferential leakage in split rings, above

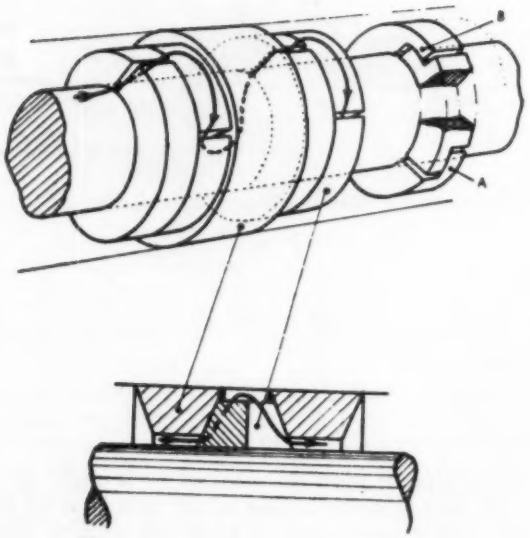


Fig. 23—Leakage through metallic packing, right

Alignment

Packings differ greatly in their adaptability to variations in alignment of the rod in the stuffing box but inevitably the packing becomes less resilient as the pressure load is increased, whereas the distortion of the machine becomes greater. In the past the open-frame type of machine has been favored because of its good accessibility, despite the lack of symmetry which, as Fig. 16 shows, results in an overturning moment on the cylinder and movement of the gland across the line of stroke. Various methods of mounting the forged-steel cylinders on the bedplates are adopted and some are better than others, but this angular change of alignment with each stroke is inseparable from this design; it can be reduced but cannot be eliminated. Good alignment, at the expense of accessibility, is possible with the A-frame design, and an alternative which combines good alignment with maximum access is becoming increasingly popular in the United States of America, where the excellent results obtained are considered amply to justify the unorthodox design, Fig. 17*b*. A related defect, from which the horizontal open-type machine is relatively free, is movement of the rod due to variations in crosshead thrust, a state of affairs which is frequently aggravated by a high value of the crank/connecting-rod ratio. Unit construction of the crosshead guide and stuffing box, Fig. 18, is the best means of ensuring that such movement as does occur has the minimum effect on alignment. An alternative, used in certain designs of diesel injection pump, is to have an intermediate crosshead linked to the main crosshead by a short strut. The importance of what may seem to be small variations in alignment is shown by an instance in which a large high-pressure compressor after a very short period of distinctly unsatisfactory service was fitted with an intermediate crosshead guided in an extension of the cylinder, Fig. 19*b*, a modification which made all the difference between failure and success so far as the gland was concerned. The need to provide for poor alignment restricts the choice of packings very considerably and may eliminate those which otherwise would be most suitable for the particular duty.

Similar alignment difficulties arise with centrifugal pumps, although for rather different reasons. Here the distortion is usually due to the connected pipework rather than to internal pressure forces. With the intention of providing maximum accessibility for the gland, the open-frame design is widely used for single-entry pumps, with the shaft mounted in pedestal bearings rigidly attached to the bedplate and the stuffing box forming part of the casing and rigidly attached to the pipework, with an all too flexible connection between the two assemblies. Expansion, associated with the pumping of hot fluids, makes conditions more severe, and good features in the design of a refinery pump are shown in Fig. 20. The casing is center-line mounted and unit construction is adopted for the shaft, bearings and stuffing box, this entire subassembly being located from the casing by a single flange and spigot. Eccentricity of the shaft or sleeves, although absent when checked statically, may develop under running conditions when poor mechanical or hydraulic balance of the impeller exists. Short, stiff shafts are essential, even at the expense of some accessibility.

Leakage

By and large, soft packings, under almost any conditions, will give the least leakage, although they may also give tight glands in the other sense of the word. When experimenting with alternative types with view to obtaining lower friction, less wear and reduced maintenance, it is important, therefore, to anticipate greater leakage and to make provision for it. Innumerable attempts to develop improved packings have been abandoned because the increased leakage in the partially developed phase looked so unpromising. When analyzing leakage it must be appreciated that not all of it travels by the same route and it is helpful to distinguish between leakage at the principal sealing face and bypass leakage, which may, in turn, be subdivided into (1) leakage around the back of the packing and (2) leakage through the packing.

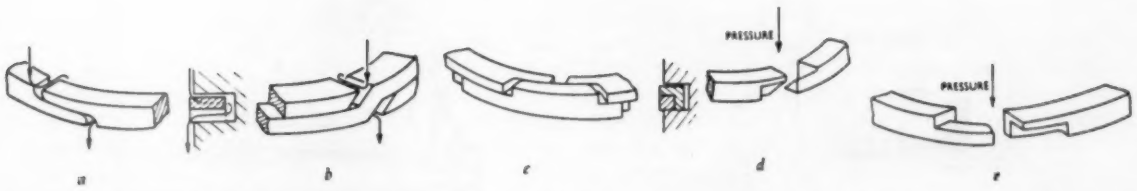


Fig. 24—Sealing of ring gaps *a, b*—Leakage by way of the back of the ring. *c*—Two-piece ring. *d*—Bevelled-step joint. *e*—Square-step joint.

Soft Packings

Soft packings remain effective only so long as they remain relatively impermeable. Most frequently grease, tallow or wax is used, sometimes assisted by graphite or mica, to seal the interstices in the fabric matrix, and the ineffectiveness of ordinary packings in, for example, refinery pumps is due to the leaching out of these substances by the powerful solvent action of the various petroleum fractions, so that the packings become permeable. Fabric proofed with rubber, either natural or synthetic, retains its impermeability but the tendency of rubber to swell and cause binding excludes its use in many instances. Metal foil is widely used as this is inherently impermeable and can be fabricated to give the necessary degree of flexibility. Greater success is sometimes achieved if the compromise solution is abandoned and separate materials are used for the two distinct functions. Thus a dam, or series of dams or cut-offs, may be formed by foil, plastic or compressed cork membranes sandwiched between resilient packing. To be effective, these impermeable membranes must be a good fit on the rod and in the stuffing box, a condition which requires good alignment. For valve stems, which are liable to stick owing to infrequent operation, greasy packing may be used even with solvents if an impermeable packing is used next to the neck bush so as to reduce percolation through the body of the packing. This practice is less effective with reciprocating pumps where the pressure pulsations make the packing "breathe."

Acid pumps are notoriously troublesome and considerable ingenuity has gone into the development of glandless types, which have their limitations. Asbestos has been practically the only available packing material until the recent introduction of the plastic polytetrafluoroethylene (P.T.F.E.). This has now been applied with considerable success as it possesses the two requirements of impermeability and chemical inertness in addition to low friction.

Packings used in reciprocating pumps are subjected to a pressure cycle, and this may lead to a type of leakage peculiar to soft packings which can, perhaps, best be described as "sponge action." Over one-half of the cycle the packing, or that portion of it adjacent to the gland, is squeezed and exudes fluid, while over the remaining half-cycle the pressure is relaxed and the packing absorbs fluid. Fluid leaks from the gland during the first half-cycle and fresh fluid flows in to replace it during the subsequent half-cycle. Dresden and Siegers (1950) have both treated a related problem in which fluid flows into and out of an absorbent packing as the pressure rises and falls, the leakage then being obtained by integrating the flow over one half-cycle. Attenuation of the pressure wave occurs with distance through the packing, but no hydraulic forces such as those shown in Fig. 2 are assumed to act on the packing. This difference is important, as the alternating compression and relaxation of the packing is the principal factor in sponge action and, from a consideration of the diagrams in Figs. 4 and 21 it will be clear that, as a result of the redistribution of compression, the packing at different ends of the stuffing box is behaving differently during any one half-cycle. The packing can behave like a sponge only when it is permeable and in most instances this type of leakage through the packing is insignificant when the packing is new but becomes progressively more important as the packing becomes exhausted.

Semi-Rigid Packings

These rely on controlled distortion to maintain the sealing action and it is common experience that their effectiveness is greatly reduced if the rings are split in order to facilitate insertion. Staggering of the joints in adjacent rings is quite ineffective because the internal channel which usually exists provides a low-resistance leakage-path through the packing, Fig. 22. One American manufacturer has now introduced a modified ring section which effectively localizes circumferential circulation to a relatively small arc. With metallic packings, rings having conical faces are frequently used, corresponding conical faces on the housings serving to

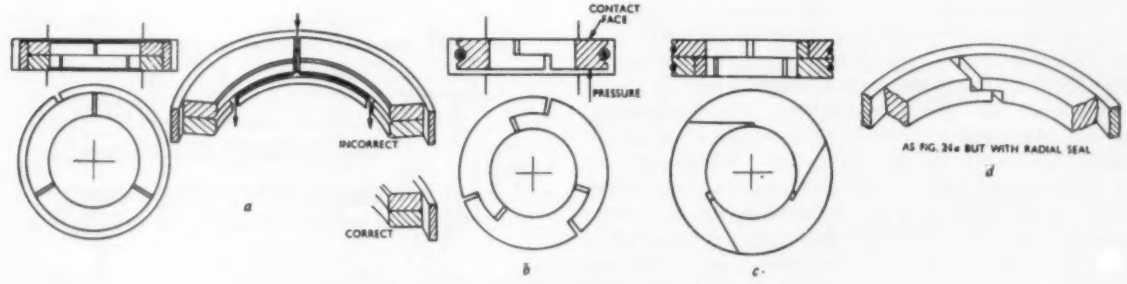


Fig. 25—Floating packings for multi-compartment glands

a—Simple segmental type. *b*—Stepped-joint type.
c—"France" type. *d*—Lapped-joint type.

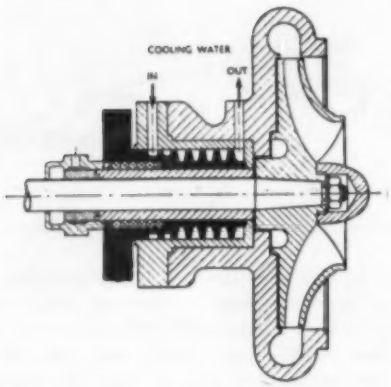


Fig. 26—Cooling-bush design of high-temperature gland

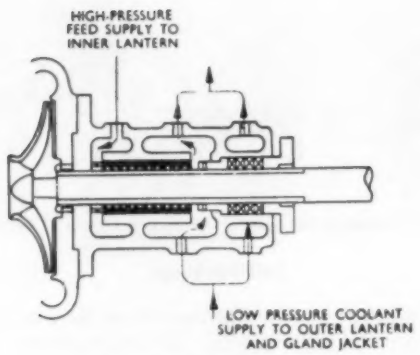


Fig. 27—Leak-in, leak-out design

contract the packing onto the rod. Unless carefully designed the packing can be bypassed completely by way of the stuffing-box wall and the bore of the housings as shown in Fig. 23. It is equally ineffective to half-lap the rings as at A, but the joint at B is satisfactory.

Floating Packings

The deficiency of the packing just considered is that the staggering of the plain joints effects an axial seal but not a radial one With any packing where the pressure has access to the back of the ring, both a radial and an axial seal are necessary. Consequently this is important with all floating packings and with piston rings, which are outwardly sealing floating packings. There has been a tendency in recent years to decry the importance of the piston-ring gap, but this is based largely on high-speed automobile practice where rapidly varying gas pressures and inertia forces may lead to intermittent loss of the seal. With the more constant conditions in the gland, leakage is mainly by way of the ring gap. The radial seal is made by contact of the ring with the groove wall and this is broken by the gap. Consequently, rings such as in Fig. 24*a* and *b*, which maintain the seal only on the face, are little more effective than rings with a plain gap. In practice the number of effective designs is limited by the additional requirements of robustness and ease of manufacture. Two-piece rings of the type shown in Fig. 24*c* are effective and a one-piece design which merits wider use, especially on the grounds of simplicity, is shown in Fig. 24*d*. With both these designs it is imperative that the ring is fitted the correct way up in the groove.

Similar features are required in the inwardly sealing floating packings used in the multi-compartment gland, where the greater freedom from dimensional restrictions permits a wider variation in design. One of the simplest, especially from the standpoint of manufacture, is shown in Fig. 25*a*. The joints in the two sets of segmental rings are staggered to provide an axial seal and the encircling ring seals the gaps radially. When used for high-pressure gas, this packing relies for its effectiveness largely on adequate pressure lubrication and a chamfer is necessary on the outer edges of the rings if a too severe oil scraping action is to be avoided. Indiscriminate chamfering of the rings on the inside edges, however, will provide a leakage path across the face of the packing as shown. The packing shown in Fig. 25*b* has fewer parts but accurate machining of the joints requires specialized manufacturing facilities, and the same is true for the type of segment shown in Fig. 25*c*. The radial seal of type *b* is effective only when the ring is fitted so that the face shown is the contact face.

High Temperature Glands

The use of a water-jacket to cool the gland is well-established practice, but this is adequate only when relatively mild temperature conditions are experienced, because the heat passing to the jacket must first be conducted through the packing. When conditions are more severe, steps must be taken to isolate the two functions of cooling and sealing, so that the packing is not in contact with a hot fluid at high pressure. The most severe conditions are probably encountered in connection with the glands of circulating pumps for forced-circulation boilers and a selection of designs for this duty will be considered in order to illustrate the principles involved. The simplest arrangement, shown in Fig. 26, is to provide a water-cooled bush in series with the packed gland, the heat being abstracted from the hot fluid during its passage through the clearance between the bush the shaft. The flow in the clearance space is limited to the leakage from the gland, so that no throttling occurs

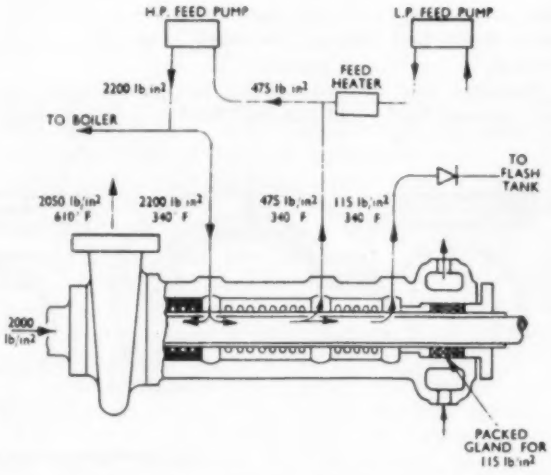


Fig. 28—Diagrammatic layout of circulating pump for 2050 lb/in.² at 610 F

and the packing is subjected to the full pressure. An alternative is to use a cold-water feed to "buffer" the gland from the hot fluid, this cold supply being at a slightly higher pressure so that there is a leak into the system instead of a leak out of the hot fluid. A tapping from the boiler-feed supply can be used for this purpose, but difficulties arise during stand-by periods when circulation must be maintained but no makeup is required. Again, the packing is subjected to the full pressure and it is somewhat unusual to find these types in use at pressures substantially in excess of 600–700 lb/in.^{2}. For higher pressures, a modification of the buffer-fluid system enables the packed gland to operate at a lower pressure than the system pressure. This is shown in Fig. 27. A clearance gland in the form of a labyrinth bush is used in series with the packed gland and through this a substantial proportion of the buffer fluid leaks to a receiver at an intermediate pressure to which the packed gland is subjected. For a given pressure drop, the flow will depend upon the clearance and the length of the bush. The need to provide for expansion renders very fine clearances impracticable and, in consequence, relatively large leakage flows must be accepted in order to achieve worth-while pressure drops. At the highest pressures the power involved in recirculating these quantities of buffer fluid is not inconsiderable, but by this means an otherwise impossible gland problem can be satisfactorily resolved. An installation in which system conditions of 2050 lb/in.^{2} and 610 F are reduced to 115 lb/in^2 and 340 F at the packed gland has been described by Clark, Rosencrants and Armacost (1943), and the arrangement is shown diagrammatically in Fig. 28. The modern tendency toward super-pressure boilers is leading to the more extensive use of glands of this type in the feed pumps also, where a split feed system is used with one or more pumps coming after the feed heaters as in Fig. 28. To keep the number of pump stages to the minimum high rotational speeds —in excess of 7000 rev/min—are being adopted and the resulting high peripheral speed of the shaft in units of large size makes the use of any form of contact seal unacceptable. These pumps are an excellent example of a design in which the requirements of the gland are dominant, and the principle of maintaining a pressure difference by means of a controlled leak has wider applications in the chemical and petroleum refining industries when exceptionally severe conditions make conventional methods impracticable.

Acknowledgment

The generous assistance of Imperial Chemical Industries Ltd., is gratefully acknowledged as is also the help of numerous firms, in particular, Bellis and Morcom Ltd., and Peter Brotherhood Ltd., who kindly loaned drawings on which some of the illustrations have been based, and of William Doxford and Sons Ltd., by whose permission Fig. 19*a* is reproduced.

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Reference Fuel Gas Available from National Bureau of Standards

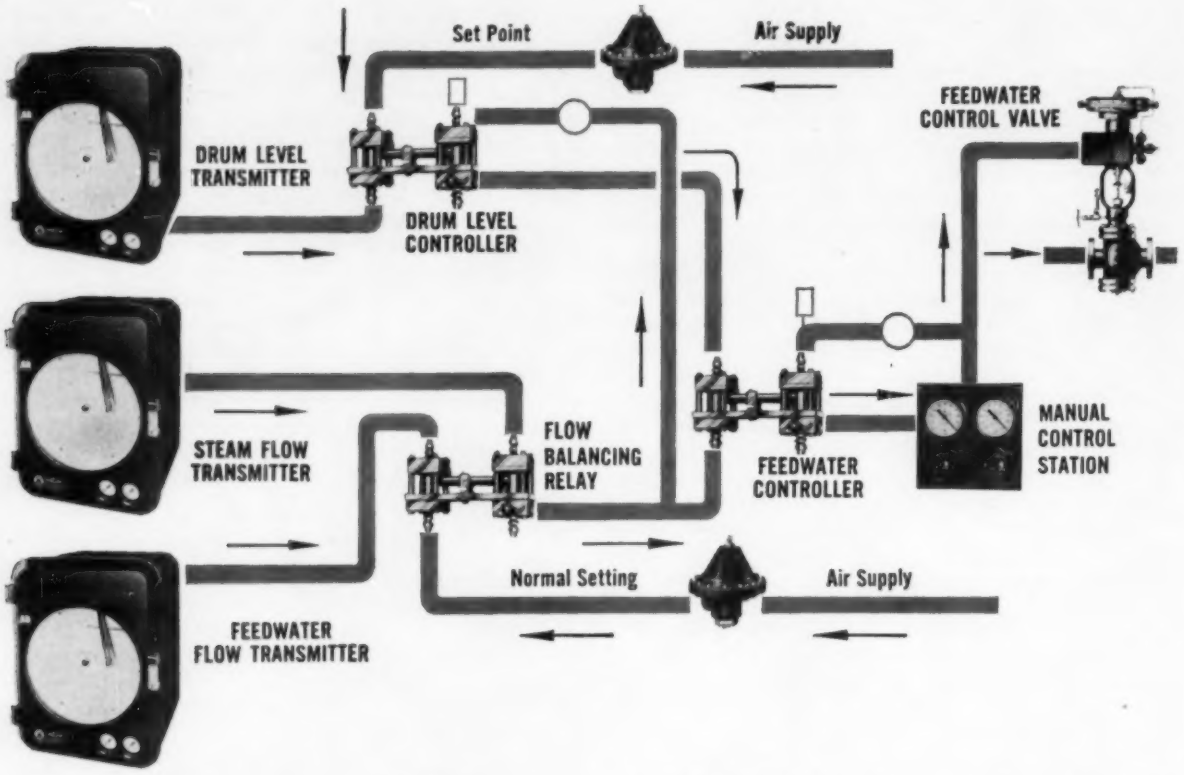
Cylinders of standard gas having a certified heating value may now be obtained from the National Bureau of Standards.[1] This recently inaugurated service is the result of a cooperative research effort by the Bureau and the Research Committee of the American Gas Association to develop a reference gas, certification procedure and calorimetric test methods acceptable to the utilities marketing natural gas as a fuel.

The tremendous increase in the use of natural gas as a fuel has focused attention on the need of a standard gas for the calibration of recording calorimeters[2] used to measure the heating value of natural gas and similar gas fuels. The accuracy of this instrument that determines the daily or hourly quality of the gas at the source is a matter of vital concern to the general consumer and the gas utility. The American Gas Association therefore initiated a general study with three specific objectives: (1) to evaluate the recording gas calorimeter, (2) to select a suitable reference gas having a heating value near that of natural gas and (3) to develop a simple standards program to insure the availability of this standard gas to industry. The Bureau actively participated in all phases of this study and, in particular, directed the technical research activities of certification.

The program now in operation is essentially the same as that previously administered by the American Gas Association. The reference gas is supplied in standard "E" cylinders containing approximately 250 SCF of gas at an initial pressure of about 1900 psi roughly sufficient for 200 calibrations. The gas has a certified heating value of approximately 1000 Btu/cf. Charges for gas, tank rental and services have been fixed so that the program is self-supporting.

[1] Request forms and additional information can be obtained by writing directly to National Bureau of Standards, Chemistry Division, Washington 25, D. C.

[2] For further information on recording calorimeters, see "Accuracy of the Cutler-Hammer Recording Gas Calorimeters When Used with Gases of High Heating Value," by J. H. Eiseman (NBS) and E. A. Potter (D. C. Gas Inspection Bureau), *J. Research NBS* 58, Vol. 213 (1957), RP 2754.



Hagan 3-Element Feedwater Control features compensated measurements and Automatic Level Monitor

Modern boiler design demands a higher degree of accuracy and response in feedwater control. The Hagan 3-element system meets this demand by providing exceptional precision of measurement and flexibility in interpreting measurement into action. Here are some reasons why the Hagan system is outstanding:

COMPENSATION—The three major measurements—drum level, feedwater flow and steam flow—can be made with Hagan Ring Balance meters compensated for variations in fluid density. This permits maintenance of accurate automatic control over a wider range of operating conditions, especially at start-up and shut-down.

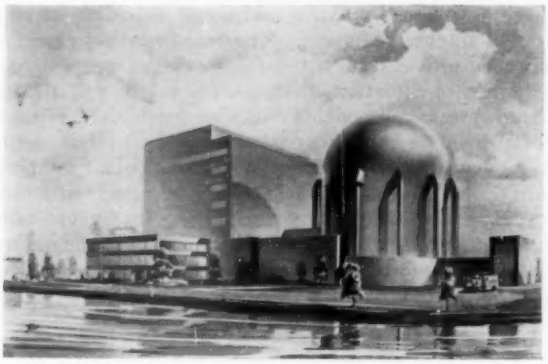
WIDE-RANGE ADJUSTMENT—The use of wide-range Hagan Ratio Totalizers as flow-balancing relays and controllers allows independent adjustment of the influence of each variable. This provides flexibility and accuracy in matching the control system to boiler and load characteristics.

AUTOMATIC LEVEL MONITOR—holds drum level at the set point by compensating for discrepancies between steam generated and steam measured. In ordinary 3-element systems the level is depressed or elevated by such discrepancies as, for instance, during blowdown or steam soot-blowing when steam is being used which is not measured by the steam flow meter.

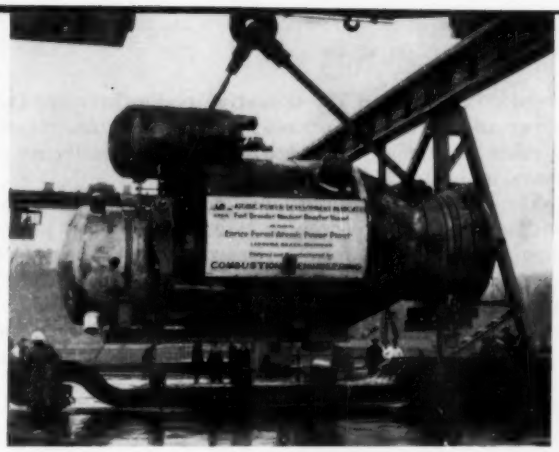
LEVEL INFLUENCE FROM BOILER LOAD—can be incorporated to hold level above normal at high loads and below normal at low loads. Write for Bulletin MSL-3, which describes such an installation on a 400 Mw station.

HAGAN CHEMICALS & CONTROLS, INC.

HAGAN BUILDING, PITTSBURGH 30, PENNSYLVANIA
DIVISIONS: CALGON COMPANY, HALL LABORATORIES
IN CANADA: HAGAN CORPORATION (CANADA) LIMITED, TORONTO



Architect's rendering of proposed Enrico Fermi plant of the APDA.



View of the loading operation as reactor vessel leaves from Chattanooga.

APDA's Fast-Breeder Reactor Vessel

SHIPMENT of the reactor vessel for the Enrico Fermi Atomic Power Plant, America's first full-scale fast-breeder nuclear power plant, marks another milestone in the development of atomic energy for peaceful purposes. The 36-ft high, 183,000-lb vessel was built from basic designs specified by Atomic Power Development Associates, Inc. Detailed design and construction were done by Combustion Engineering, Inc., at its Chattanooga Division.

Description of Vessel

The reactor vessel is the most complex stainless steel pressure vessel of its size ever built. It is made up of four separate cylindrical vessels welded together to form a unit assembly. The four major components of 36-ft high reactor vessel are the lower vessel, the upper vessel, the conical transition section and plug container, and the transfer rotor container.

The design and manufacture were greatly complicated by the eccentric relationships of these components and the severe thermal and nuclear radiation conditions anticipated in operation.

The lower reactor vessel, which is 9 ft. 6 in. in diameter, houses the baffling and thermal shielding necessary for protecting the vessel from radiation damage and thermal stresses. It also distributes the inlet coolant flow through the nuclear fuel, the plutonium breeder blanket, the control rods, and the shield. The lower reactor vessel contains the support plates which support and locate 673 eight-ft long, $2^1/_2$-in.-square tubes containing the nuclear fuel, breeder blanket, and nuclear control and safety rods. These extend to a point just above the junction of the lower reactor vessel with the upper reactor vessel.

The upper reactor vessel, which is 14 ft, $6^1/_2$ in. in diameter, houses the fuel handling device, the holddown plate, and the eight safety rods. The main sodium coolant discharge is located in this section. The upper reactor vessel will be operated at approximately atmospheric pressure, with an inert gas atmosphere maintained above the sodium level.

The conical transition section and the plug container is a single unit welded to the upper reactor vessel. This section houses and supports the rotating plug, which in turn houses and supports the fuel handling device, the holddown drive mechanism, and control and safety rod drive mechanisms.

The transfer rotor container houses the transfer rotor which provides for lateral transfer of the fuel and blanket sub-assemblies to a position where they can be removed from the reactor vessel. It also provides the temporary decay storage for irradiated sub-assemblies immediately after removal from the reactor.

The reactor is supported by twelve flex plates which are attached to the bottom of the upper reactor vessel and bolted to foundation plates located below the bottom head of the lower reactor vessel. These flex plates are 7 ft, $4^1/_2$ in. long and are designed to allow for the thermal expansion of the vessel during operation while maintaining its correct horizontal position under all operating conditions. These flex plates will support a total operating weight of approximately 600 tons.

Design of Reactor Vessel

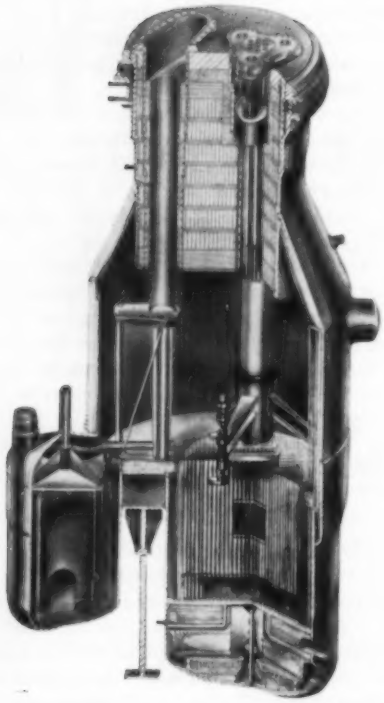
The reactor vessel is designed for 50 psig at 1000 F at the outlet for the upper reactor vessel, and 110 psig inlet pressure at 650 F for the lower reactor vessel. Because of these low pressures it was possible to make use of a thin-wall reactor vessel design. The wall thicknesses for the APDA reactor vessel were determined from the applicable requirements of the ASME Boiler and Pressure Vessel Code, Section VIII, Unfired Pressure Vessels. In most areas, the vessel wall is heavier than required by code standards because of the fabrication handling problem associated with a thin-walled vessel of such large diameter.

The design considerations of this vessel were: pressure, steady-state temperature, temperature transients, internal weights and external reactions. Areas for which no specific design procedure is presented in the Code were investigated by advanced methods of stress analysis. Such areas include the support structure for the fuel and

breeder blanket and the transition region between the upper and lower reactor vessels. The transition region is made additionally complex because the transfer rotor container is attached to this region.

The walls of the reactor vessel will be lined internally with many layers of concentric stainless steel shield plates ranging in thickness from $^1/_8$ in. to one in. Their total thickness amounts to $4^1/_2$ in. in the upper reactor vessel and 6 in. in the lower reactor vessel. Two circumferential rows of solid stainless steel bars, placed between the reactor core and adjacent to the lower reactor vessel shielding plates, provide additional shielding in this area of intense radiation. The reactor vessel shielding has been provided for a two-fold purpose: primarily for reduction of radiation effects on the vessel wall and external system components which are within the primary plant shielding and secondarily, to serve as a buffer zone to prevent a sudden change in sodium coolant temperature from being experienced by the vessel structure. Rapid temperature changes of the sodium coolant of as much as 300 deg F in one second can occur in the sodium leaving the reactor due to the rapidity of shut down.

FACT SHEET—APDA REACTOR VESSEL



Sectional drawing of the APDA reactor vessels shows in upper section the plug container and conical transition sections; immediately below is the area known as the upper reactor vessel; to the left of the main body is the transfer rotor container and immediately below the main body is the lower reactor vessel.

1. WEIGHT
   Vessel shell without internal components—182,099 lb.
2. WEIGHT
   Rotating plug, thermal baffling, internal shielding holddown mechanism, support plate assembly, fuel sub-assemblies, blanket sub-assemblies, sodium coolant and additional component parts—approximately 350 tons.
3. WEIGHT
   Rotating plug and lower 18 in. of shielding (to be shipped separately from vessel)—216,915 lb.
4. MATERIAL
   900,000 lb (450 tons) of stainless steel and 65 tons of carbon steel, plate, bar and forging was ordered for fabrication of the vessel, internal shielding, rotating plug and other structures.
   Every square inch of plate used in the vessel was ultrasonically inspected for flaws.
   Vessel shell: Up to 2-in.-thick Type 304 stainless steel.
   Rotating plug cover plate—12 in. thick, 116 in. in diameter, made of carbon steel forging.
5. DIMENSIONS
   Overall height of vessel with rotating plug—37 ft, 8 in.
   Diameters—Lower reactor vessel—9 ft, 6 in. (outside)
   Upper reactor vessel—14 ft, $6^1/_2$ in. (outside)
   Plug container—9 ft, 6 in. (outside)
6. MANUFACTURING TIME
   28 months; work done on a three-shift, six-day-week basis.
7. WELDS
   Total feet of weld in vessel—2200.
   Total feet of weld in the shielding, rotating plug, and miscellaneous structures—5900
   All major welds were radiographed, using various types of x-ray equipment, including 15,000,000-volt betatron.
   All welds were tested for flaws by either a dye-penetrant or magnetic-particle test.
8. MANUFACTURING
   Combustion Engineering, Inc., Chattanooga Division.

Static Automatic Control for Electrical Precipitators

By LARRY L. LITTLE

Western Precipitation Corp.

The desire for a complete and reliable automation system for electrostatic precipitators has prompted considerable investigation into ways and means of obtaining it. The author advances the arguments for one such system which employs static solid state components in the control circuit to produce among other things a maximum of reliability.

AUTOMATIC power control for electrical precipitators serves several desirable functions. The most obvious function is to obtain highest average collection efficiency; however, in many cases an equally important advantage is maximum collection efficiency with minimum increase in stack emission during peak dust loads due to cyclic or transient plant process changes. Other important features of an automatic power control include a reduction in down time and savings in maintenance and operator labor. Previous objections to automatic precipitator controls such as circuit complexity, component wear, failure, inadequate range of control and high capital investment have been eliminated or minimized in a new control unit.

The need for an automatic control for an electrical precipitator is most evident in applications where the gas characteristics and the properties of the suspended particles change rapidly or change over a wide range. Even where the changes require several hours, or where the precipitator voltage or current needs to be altered by only a few per cent, however, an automatic control may be well worth its cost.

Theoretical Basis of the New Control

Plots of precipitator efficiency versus power input, in general, yield curves similar to Fig. 1. The characteristic initial rise in efficiency with increased power input, the point of maximum efficiency with optimum power input and the eventual reduction in efficiency due to the effects of heavy sparking as power input becomes excessive seem universal. Similar curves have been developed by others (1)[1] for efficiency versus sparking rate. Alteration of the electrical properties of the gas and particulate matter will change the power and the sparking rate at which maximum efficiency occurs (2, 3).

Figs. 2 and 3 are reproductions of oscillograph traces of the voltage across the saturable core reactor connected in series with the precipitation transformer primary winding on a dry process cement application. The regular portion of the traces show the reactor voltage without sparking, while the random peaks show transient reactor voltage due to precipitator sparks. Oscilloscope calibration is the same for the two figures.

Fig. 2 shows voltage surges resulting from sparks of high energy value associated generally with collected material having moderate resistivity, while Fig. 3 shows voltage surges resulting from fine low energy sparks associated with collected material having higher electrical resistivity.

In general, when the electrical resistivity of the collected material is high, maximum efficiency is associated with high sparking rates, typically several hundred per minute. When the resistivity of the collected material is moderate, somewhat lower sparking rates of 10 to 100 sparks per minute result in maximum collection efficiency, (1). In addition, when the electrical resistivity of the collected material is high, the sparks are very fine and dissipate very little precipitator power. These sparks individually cause little change in precipitator

[1] Numbers in parentheses apply to similarly numbered entries in a list of References at the close of the article.

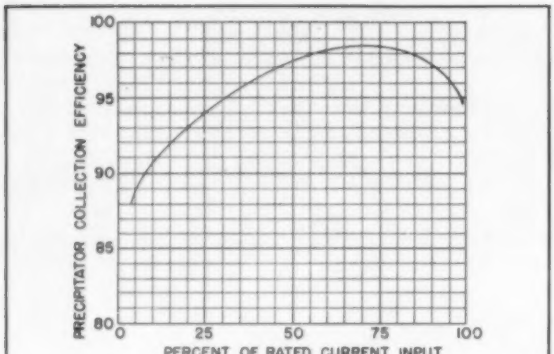


FIG. 1

PRECIPITATOR COLLECTION EFFICIENCY VERSUS CURRENT INPUT

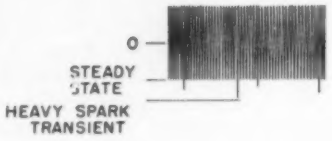


FIG. 2

INTENSE SPARKS* IN PRECIPITATOR COLLECTING MATERIAL HAVING A MODERATE ELECTRICAL RESISTIVITY

*OSCILLOSCOPE CONNECTED ACROSS REACTOR AS SHOWN IN FIG. 4.

voltage and current and erode very little material from the collecting surfaces. On the other hand, when the electrical resistivity of the collected material is lower, the spark intensity is higher. Each spark discharges the precipitator more than when the resistivity is higher and more erosion of the collected material is produced by the higher energy spark.

As the resistivity of the collected material increases, the desired number of sparks per minute for maximum collection efficiency increases and the energy value of each spark decreases. As the resistivity of the collected material decreases, the desired number of sparks per minute decreases while the average energy of the individual sparks increases. It can be seen that while the desired spark rate and spark energy each vary widely, the product of spark rate and spark energy, i.e., the average sparking current, tends to remain constant. This constant average value of sparking current is the basis of the new automatic precipitator power control.

Mode of Operation

When a spark occurs between an ionizing wire and the grounded collecting surface in the precipitator, there is an instantaneous reduction in precipitator voltage and hence rectifier output voltage. From Fig. 4 it can be seen that this spark, by reducing the precipitator impedance and hence the rectifier voltage, causes a corresponding temporary increase in voltage drop across the saturable core reactor impedance in series with the incoming power to the precipitation transformer and rectifier. The amount and duration of the voltage increase across the series connected saturable core reactor depends upon the magnitude and duration of the current discharged through the spark, i.e., spark intensity. These voltage changes across the saturable core reactor, when added together in a suitable summing circuit, produce a signal which is proportional to spark rate and spark intensity or to the average sparking current.

This signal output from the summing circuit, which is proportional to the average sparking current, is compared with the desired average sparking current level. The difference between the desired level and actual level is amplified by a silicon transistor and used to drive a magnetic amplifier. The magnetic amplifier serves to reverse the sense of the sparking current signal as well as to increase as the sparking current increases. This signal from the transistor is applied to the magnetic amplifier in the proper polarity to reduce the output of the magnetic amplifier as the input, i.e., sparking current, increases. The magnetic amplifier in turn supplies saturating current to the saturable core reactor which controls the power to the precipitator. The effect of the entire control circuit is to increase precipitator current as sparking current diminishes and to decrease precipitator current as sparking increases. By this method precipitator power is maintained at a level which gives optimum sparking current.

Design Features

This automatic power control unit has been found to maintain optimum power input to the precipitator even with wide changes in the electrical properties of the gas and particulate matter entering the precipitator. When changes occur in the precipitator which alter spark intensity, the control automatically alters spark frequency to meet the new conditions. No manual readjustment is necessary to obtain full range of power control.

The control circuit is completely static in operation. There are no moving parts to wear and no filaments to burn out. All the parts in the control circuit have the long trouble-free life expectancy associated with transformers, silicon devices and oil-filled capacitors. Conservative design factors are used throughout the design to minimize the possibility of failure due to overload. The small parts which might be subject to mechanical or corrosion damage are enclosed in a metal container and sealed in special potting compound.

Testing procedures are very simple and straightforward due to the basic simplicity of the circuit and to the arrangement of the circuit in subassemblies or modules.

Satisfactory automatic power control operation presupposes careful design of the electrical precipitator, taking into account the mechanical and electrical properties of the gas and particulate matter entering the precipitator. Collecting surface designs which have stable electrical characteristics and means to reduce particle re-entrainment, rappers with adequate, but not excessive impact to properly clean the discharge and collecting surfaces, discharge electrodes which supply good ionization, and even gas distribution, are vital to satisfactory precipitator performance. These features, together with modern reliable automatic control, insure maximum collection efficiency with minimum maintenance and adjustment year after year.

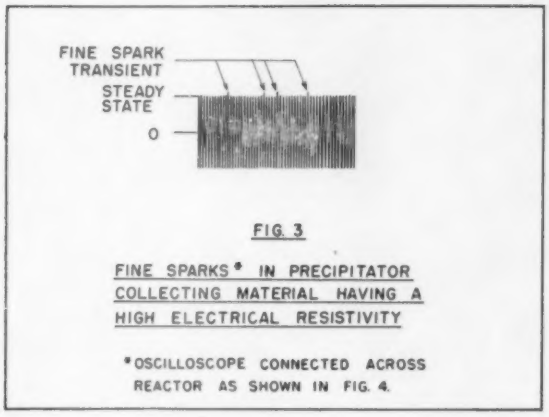


FIG. 3

FINE SPARKS* IN PRECIPITATOR COLLECTING MATERIAL HAVING A HIGH ELECTRICAL RESISTIVITY

*OSCILLOSCOPE CONNECTED ACROSS REACTOR AS SHOWN IN FIG. 4.

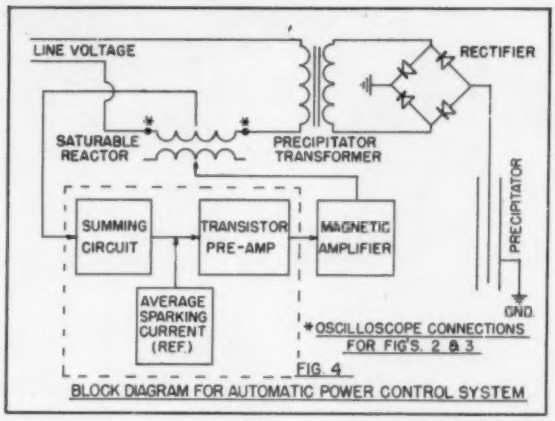


FIG. 4

BLOCK DIAGRAM FOR AUTOMATIC POWER CONTROL SYSTEM

Performance

Portable automatic controls of the type described above were installed on several precipitators during the year of 1956. On a typical fly ash precipitator in a large generating station the automatic control maintained the average power input to the precipitator at approximately $3^1/_2$ times the normal input power with manual resistance control. The current input to the precipitator varied from 120 ma to 280 ma during the test period and the voltage varied from 35 kv to 49 kv. On the usual manual setting, the current varied from 65 ma to 75 ma and the voltage varied from 33 kv to 37 kv. From Fig. 1 it is apparent that substantial improvement in collection efficiency was obtained since the automatic control strives for maximum power but prevents excess sparking.

Automatic controls were installed on two, series flow, pipe-type precipitators in a metallurgical installation. With automatic control the precipitator voltage varied from 52 kv down to 5 kv. The primary current was varied automatically from 10 to 40 amperes. On the usual manual settings the secondary voltage varied from 46 kv down to 5 kv. Due to extreme changes in operating conditions, the normal primary current input with manual control was maintained at less than 10 amperes to prevent overload trip-out. Precipitator dust loss through the precipitator was 0.0279 grains per cubic foot with automatic control and 0.104 grains per cubic foot with manual control operation. Following the demonstrations and tests at this installation, all of the precipitators were equipped with automatic controls. These controls have been in operation approximately one year and are performing very satisfactorily.

The precipitators cleaning the effluent gas from 5 open hearth steel furnaces were equipped with a total of 15 automatic controls. Some of these controls have been in operation approximately one year. All of these controls are now operating effectively over the extreme range of conditions found in this application.

Approximately 100 automatic control units are currently being installed with new precipitators in fly ash, cement, chemical and metallurgical installations.

REFERENCES

(1) "An Automatic Voltage Control System for Electrical Precipitators," by H. J. Hall, *Communication and Electronics*, May 1954.

(2) "Operation of Cottrell Precipitators, Effects of Moisture and Temperature," by Wayne T. Sproull, Yoshinao Nakada, *Industrial and Engineering Chemistry*, vol. 43, June 1951.

(3) "Collecting High Resistivity Dusts and Fumes," by Wayne T. Sproull, *Industrial and Engineering Chemistry*, vol. 47, May 1955.

A Rare Earth Now Available

Dysprosium metal, one of the rare earths, is now being produced by Nuclear Corp. of America in commercial quantities, the company recently announced.

Available only in tiny quantities heretofore, the "new" metal probably will find applications in the construction of nuclear reactors, in magnetic alloys for use in electronic devices and in other fields to be determined by further research, a company spokesman said.

Nuclear Corp. said it is producing dysprosium of 98 to 99 per cent purity in lump and ingot form in grams up to pound lots.

The metal's neutron cross-section is 1100 barns, which means that it has a great ability to absorb neutrons. This plus its relatively high melting point of 1400 C suggests use in nuclear-reactor control rods.

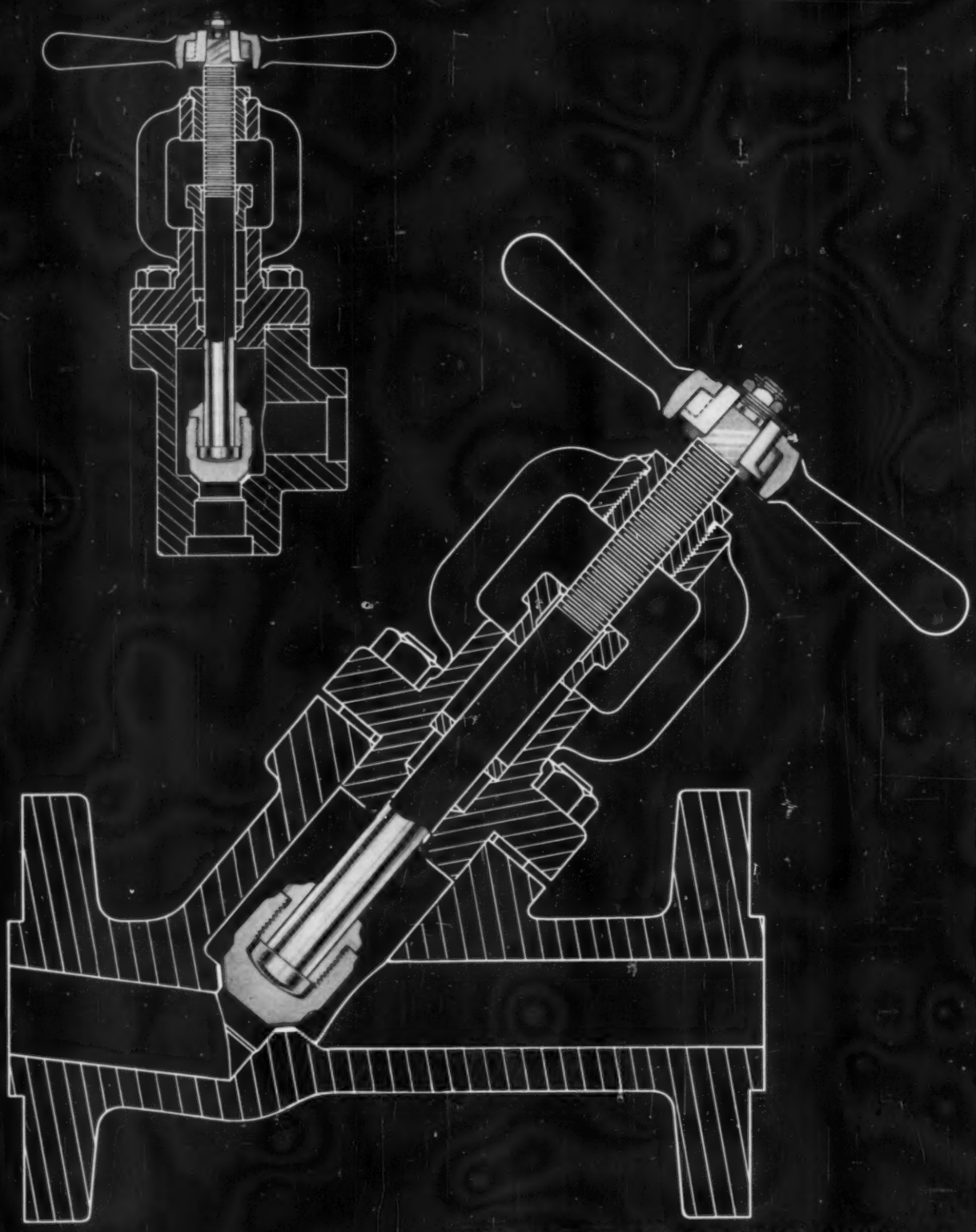
A group of scarce elements with similar properties, the rare earths occur naturally only as oxides. They always are found mixed together and with other elements. Separation and reduction of the oxides to pure metal was one of the most challenging chemical problems.

Safety and Relief Valves

A new test code for safety and relief valves has just been published by The American Society of Mechanical Engineers. The new instructions provide a recommended test procedure and instrumentation for testing valves handling steam, gases (including air) and liquids "in such states that flashing will not take place." Spring-loaded, weight-loaded and pilot-operated valves are included but only for tests at atmospheric pressure.

The purpose of the code, according to the ASME committee, is to establish rules for safety and relief valve tests to determine one or more of the following: relieving pressure, relieving capacity at the relieving pressure, theoretical relieving capacity, coefficient of discharge, start-to-leak pressure, opening and closing pressures, seal-off pressure, blowdown, reproducibility of valve performance and mechanical characteristics of the valve as determined physically.

Besides the usual explicit directions for setting up tests, making arrangements between contracting parties and delivering a complete report, the code is illustrated.



EDWARD FORGED STEEL BLOW-OFF VALVES

What's New from Edward Valves

New Products . . . Problems and Solutions. . . Information on Steel Valves from Edward, Long-Time Leader in the Field!

Edward Drop Forged Blow-off Valves

. . . the first—and still the newest!

Fifty years ago, when boilers operated at 250 lb sp the first really efficient means of blowing-down was with "new" Edward blow-off valves. Today, with pressure in the 1800-2600 lb range and with water temperatures sometimes exceeding 600 F, corresponding improvements in Edward blow-off valves have been effected. For example, today all Edward blow-off valves have integral Stellite seats and Impactor* easy closing handles.

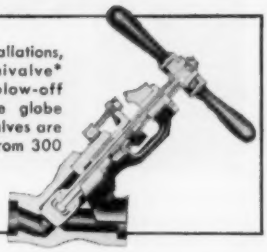
All over the world, high temperature-pressure steam power stations rely on these Rockwell-built Edward blow-off valves. The reasons are basic. Edward valves are simple and rugged in design, with no "trick construction." They operate easily, stay tight, need very little maintenance. If parts replacement is ever necessary, repairs are easier and less expensive than with any comparable valve.

The ASME Power Boiler Code requires steel valves wherever boiler drum pressure is above 200 psi . . . but, because blow-off service is severe at any pressure, operators of lower-pressure plants also have found it most economical to standardize on Edward valves.

Catalog 12-D2 tells the full story.

Recommended practice: Open valve *nearer* boiler first. Then slowly open other valve. After blow-off, close second valve first; then valve nearer boiler. This places greatest wear on blowing valve, assures a tight valve next to boiler, facilitates repair without a boiler shut-down.

Popular for high pressure installations, the Edward welded-bonnet Univalve* meets Code requirements for blow-off service and is ideal all-purpose globe valve as well. Edward blow-off valves are available in all pressure classes from 300 lb to 2500 lb in angle or straight-way styles, with flanged or welding ends.



Premium Features of Edward Blow-off Valves

- INTEGRAL SEAT of chromium-cobalt-tungsten alloy. Applied directly to body in a continuous ring, then machined in same set-up with bonnet guide bore for perfect alignment. Highly resistant to abrasion and corrosion.
- HARD-FACED DISK—Stellited for maximum resistance to abrasion and corrosion. Edward swivel-disk construction allows stem to lift vertically away from seat. No grinding or galling.
- POSITIVE BACKSEAT isolates packing from the high temperature zone. Line-contact backseat seals EValpak* valve packing from line pressure, permits repacking under pressure.
- IMPACTOR HANDLE, an Edward exclusive, is standard equipment. Several times more effective than ordinary handles or handwheels in opening or closing valves.

EDWARD VALVES, INC.

1206 West 145th Street, East Chicago, Indiana

Subsidiary of

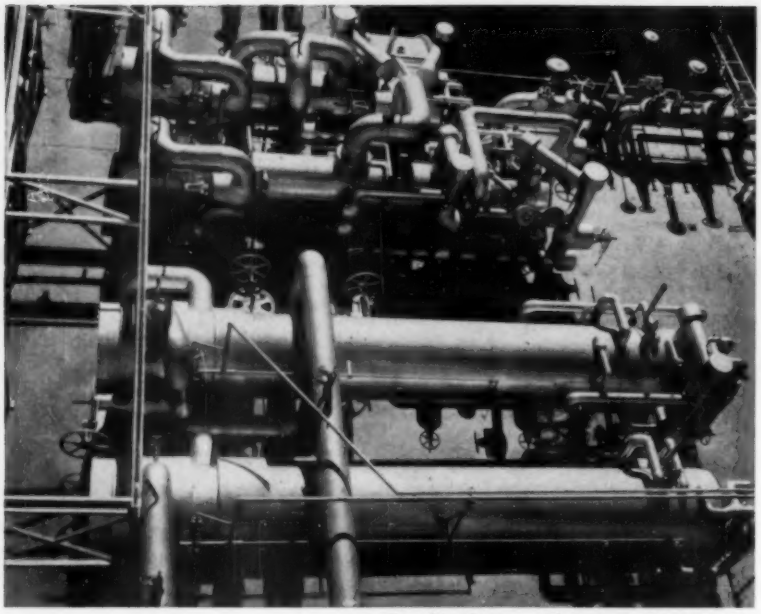
ROCKWELL MANUFACTURING COMPANY

Represented in Canada by

LYTLE ENGINEERING SPECIALTIES, LTD., 360 Notre Dame St. W., Montreal 1, Que.

Edward builds a complete line of forged and cast steel valves from ⅛" to 18"; in globe and angle stop, gate, non-return, check, blow-off, stop-check, relief, hydraulic, gage and special designs; for pressures up to 10,000 lbs; with pressure-seal, bolted, union or welded bonnets; with screwed, welding or flanged ends.

*T. M. Reg. U. S. Pat Off.



Consulting Engineers, Ebasco Services Incorporated

The modern, efficient equipment shown here was designed and built by the Yuba Heat Transfer Division for the Collin Station of the Texas Power & Light Co. Serving a 110,000 KW turbine are five Yuba feedwater heaters, of which four are shown here. Two high-pressure heaters are designed for 2600 psi on the tube side with 6500 and 6750 sq. ft. of surface. Three low-pressure heaters are designed for 400 psi on the tube side and have 2850, 3165 and 4670 sq. ft. of surface.

Yuba designed and built the first all-welded feedwater heaters, in service for the last year and a half. The weld technique was used in the tube bundles, shells and Multilok Closures. These all-weld heaters have been so successful they have started an industry trend.

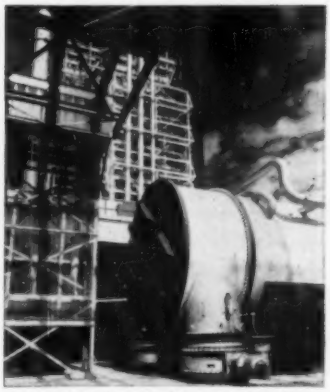
In the manufacture of condensers, Yuba is building units having 165,000 sq. ft. of total surface. The designs are readily adapted to condensers of even greater capacity as required for larger turbines.

Yuba evaporators have achieved boiler make-up water purity of less than .2 ppm. Yuba can *guarantee* purity of .25 ppm solids content per 3000 ppm shell concentration.

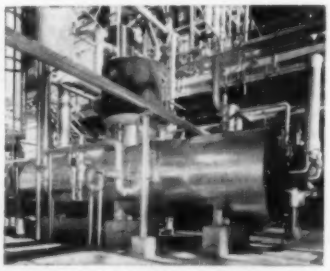
Engineering accomplishments such as these keep the power industry up-to-date. There's progress in power through progress in heat transfer equipment.

YUBA

EQUIPMENT will still be UP-TO-DATE TOMORROW because IT'S AHEAD TODAY



Twin-unit surface condenser has 90,000 sq. ft. of surface. Operates at 26.52 inches of vacuum with oxygen content in the condensate less than 0.01 cc per liter.



Bubble-tray evaporator produces 34,000 lb./hr. of vapor well below 1.0 ppm total solids with 3000 ppm concentration.

YUBA HEAT TRANSFER DIVISION

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American Power Conference in Review—II

LAST month's issue of COMBUSTION carried a major portion of the American Power Conference papers held to be of value to the readers of this magazine. There were, however, so many that in the interest of space the complete meeting report was divided into two parts and the following was held over for this issue.

Gas Turbines

J. E. Parker, Union Carbide Chemicals Co., opened the Gas Turbines session with his paper "A Gas Turbine in the Industrial Steam-Power Cycle." The gas turbine, said Mr. Parker, can be effectively utilized in an industrial plant only if conditions are right and it is properly applied. However, a solution that is good and applies in one plant may not necessarily be the proper solution for another plant.

The Texas City Plant of the author's organization was initially placed in operation in 1941. Steam and power requirements continued to grow and authorization was given in 1956 for an additional 200,000 lb per hr, 1000 psi boiler and a 7500 kva, 13,800 volt turbine generator. It was planned to make a unit installation of these two items as part of their No. 2 Steam and Power Plant which now includes two 200,000 lb per hr, 1000 psi boilers and one 6250 kva, 13,800 volt turbine generator. During this same period, another problem had become more acute—the problem of adequate compressed air. The air shortage was being partially met by portable compressors but it was not until it was found that the number of portable compressors in continuous service had risen to nearly 30, and that additional large compressed air requirements were developing, that it was decided some permanent solution would have to be achieved.

Motors or steam turbines as compressors could not be justified because of the adverse effect on the plant steam and power balance, and because of the high utilized investment costs for steam and power. This narrowed the consideration to reciprocating gas engine-driven compressors and the centrifugal gas turbine-driven compressor.

At this same time, it began to appear that the new boiler installation would be inadequate to meet the new load demands. Therefore, the possibility of utilizing the gas turbine exhaust heat for steam generation such as in a waste heat boiler gained prominence. With the heat available in the exhaust gases, as much as 35,000 or 40,000 lb per hr of steam could be generated in a waste heat boiler. However, the most effective utilization would be achieved by generating low pressure and low temperature steam while the plant need was for steam of higher pressure and temperature.

A solution presented itself because of the basic feature of gas turbines generally considered to be a handicap since it causes the low thermal efficiency—that is the high excess air in the exhaust. Since the power developed by the turbine is dependent, among other things, upon the mass gas flow and also since the temperatures through the machine have to be controlled in line with the capabilities of the metallurgy, the gas turbine operates with about 500 per cent excess air as compared to that required for the combustion of fuel. The available oxygen in the exhaust is about 18–18½ per cent of the total flow which is only slightly lower than that in the air. Therefore, the exhaust stream of the gas turbine instead of being exhaust gases, as we normally think of them in connection with engine or boiler exhaust, is actually highly heated air. The author remarked that his concern refers to the gas turbine they are installing as a combination forced draft fan and air preheater.

On this basis between 250,000 and 300,000 lb per hr of 1000 lb steam at 850 F could be generated from the new boiler with only a 10 per cent increase in cost to convert the boiler design to these new conditions. The primary change required was to enlarge the gas passages to accommodate the larger flue gas flow. Further, the boiler was originally planned to have an air preheater and since this was no longer necessary it was replaced by an economizer section. With the air preheater, it was found to be more efficient to go to 70 lb deaeration pressure. However, with the conversion to an economizer section, the deaeration pressure was dropped to 10 lb to give a lower temperature water to the economizer, and therefore permit a greater heat recovery and lower exhaust flue gas temperature. With these changes the boiler was rerated for 250,000 lb per hr when operating on gas turbine exhaust. This additional high pressure steam capacity made it possible to increase the size of the steam turbine generator from 7500 kw to 9375 kw.

The high combination efficiency of boiler and gas turbine that resulted from this also reflected another advantage to the gas turbine—reduced operating cost. An estimated operating cost per thousand cubic feet of air produced is 26 per cent lower for the gas turbine than for a gas engine reciprocating compressor. There will also be reduction in steam costs due to the higher efficiency and to the reduced operating personnel requirements per unit of steam since the same personnel can

operate the larger boiler and gas turbine combination as would be required to operate the boiler alone. There will also be an auxiliary power saving since it will not be necessary to operate a forced draft fan for combustion.

The simplicity and compactness of the gas turbine, as well as the high capacity per unit, gave another savings in installation cost. It was estimated that per unit of capacity, the purchase and installation cost for the gas turbine would be 14 per cent less than that for a reciprocating installation. It is interesting to note that this size advantage which reflects the 14 per cent reduction in installation cost also appeared earlier as a disadvantage when considering the capacity in a single unit.

W. D. Marsh followed with the paper "How to Integrate Gas Turbines with Other Generation in an Economical Utility Generating System." The purpose of this paper was to show how economic conditions governing the use of peaking generation have changed radically and, second, to describe methods for integrating gas turbines with other generation on a utility system for efficient handling of peak loads.

The author then defined the hypothetical case for peaking and examined it against the 30-year period from 1925 to 1955. At the beginning of this period a system that always installed the best units available in each year would have as its best heat rate about 15,000 Btu per kwhr. Looking backwards, its oldest unit would be upwards of 40,000 Btu per kwhr heat rate. The average of all units would be about 25,000 Btu per kwhr. Could this system in 1925, posed Mr. Parker, justify buying a peaking unit? Put another way, could it afford *not* to buy another "best available" unit, having a heat rate 10,000 Btu per kwhr better than the system average? The answer, of course, is "No"—and in 1925 utilities were not buying peaking units.

By the end of the period, in 1955, the situation became somewhat different. By then, some of the generation existing in 1925 has been retired. How much need not be considered since the system had grown to eight times larger, and all of the 1925 capacity would represent only a few per cent of the total. The best unit on the system had a heat rate of about 9000 Btu per kwhr. Because most of the generation had been added in the later years, the system average was slightly over 10,000 Btu per kwhr. Can such a system in 1955 justify a peaking unit, the author asked? Can it afford not to add a "best available" unit? The answer was not so clear. The best available unit was only about 1200 Btu per kwhr better than the system average. Recall that in 1925 there was a 10,000 Btu per kwhr difference. The economics obviously had changed, with the result that in some cases a small amount of peaking capacity may be justified with $20 or $30 per kw reduction in investment; and as the heat rate curve flattens out beyond 1955, which it has done and systems continue to grow, peaking generation becomes more and more economical.

This trend will become even more pronounced in the future by merely assuming a system whose load doubles each ten years and who always adds the best units available. A 1955 curve, which the author displayed, showed that 100 per cent of the capacity was 14,600 Btu per kwhr, or better; 80 per cent is 11,000 Btu per kwhr or better. As a matter of fact, this curve was quite realistic because some of the large utilities in this country had even a greater proportion of efficient units in 1955.

W. H. Ferguson and **R. E. Strong,** Westinghouse Electric Corp., wound up the session with "Gas Turbines as an Economic Generating Source." The gas turbine in combination with the steam cycle, stated the authors, is potentially the next big step in the improvement of heat rates in base load power generation plants. A gas turbine plant may be located conveniently close to the load to reduce transmission investment and losses because it does not need a supply of condensing water. Sound levels comparable with those of any other prime mover can be achieved with noise suppressers in the inlet and exhaust ducting. Because neither boiler nor condenser is used and because smaller buildings and foundations are needed, a gas turbine plant can be installed at a capital cost saving of 20 to 25 per cent as compared to a steam plant of the same net output rating.

Operating labor costs are low in attended plants and can be reduced almost to zero in an unattended, remote-operation plant. The gas turbine plant in stand-by service requires no attendance, burns no fuel and suffers essentially no deterioration; it has the ability to start and pick up load quickly. A gas turbine has considerable overload capability. In peaking service it can carry loads 10 to 15 per cent above its continuous rating. At low ambient temperatures its maximum capability reaches 150 per cent of the nominal rating.

The fuel consumption of a gas turbine plant without heat exchange equipment is relatively higher than that of a base load steam plant. Heat rates range in the order of 18,000 to 19,000 Btu per kwhr at nominal full load and up to 28,000 Btu per kwhr at 50 per cent load. Under cold weather conditions when the air density is higher, the output rises well above the normal rating and the heat rate falls to about 16,000 Btu per kwhr.

Water Treatment

I. M. Abrams, B. N. Dickinson and **M. Roberts,** Chemical Process Co., on the subject "The Effect of Dissolved Oxygen on Weak and Strong Base Anion Exchangers," made the statement that many studies have been made on the causes for anion resin instability leading to two recognized causes for resin deterioration: Organic fouling and chemical attack. Fouling, the authors held, is generally associated with strong base resins and has been attributed to naturally occurring organic substances usually found in surface waters. Chemical attack may occur either on the matrix of the resin or on the amine groups attached to the resin. This attack may be either hydrolytic or oxidative. This paper was concerned with the latter.

The effect of dissolved oxygen on strong base resin performance has been a subject of diverse opinions. The authors intended by this paper to shed some new light on the apparent difference of opinion with the hope that the information may aid in prolonging the life of anion exchange resins. Both the uptake and the effects of dissolved oxygen on weak and strong base anion exchangers were studied. In addition to capacity measurements, changes in resin bed volumes and in rinse requirements were determined.

The results presented here verify those reported by Peents and Wirth. Whereas Wirth's experiments were carried out at 150 F, at which temperature Type II resins are unstable, the tests reported here were done at tem-

peratures below 115 F, where temperature alone does not affect the resin. At no time were the columns allowed to dry out. Therefore these results, the authors felt, were not subject to the criticisms directed at previous work.

The experiments described were also carried out with anion exchange resins of other manufacturers. Although the data with these other resins were not reported here, the conclusions the authors stated with regard to oxygen uptake, resin volume changes and capacity decline corresponded very closely with those presented.

F. B. Wink, Niagara Mohawk Power Corp., and **D. Miller,** The Permutit Co., collaborated on the report "Comparison of Mixed Bed and Four Bed Demineralizers."

Recently, the report declared, considerable interest has been shown in the use of four bed demineralizing systems, comprising primary cation exchanger, primary anion exchanger of either the strongly basic or weakly basic type, secondary cation exchanger and secondary anion exchanger of the strongly basic type.

In general, equipment manufacturers have indicated that effluent conductivity of four bed systems will be somewhat higher than from mixed bed systems, but usually about the same silica has been predicted from either type of system.

The mixed bed demineralizing system at the Huntley Station of Mr. Wink's company has been operating since January 1954, while the four bed system went into service in September 1956. Careful records have been kept on the operation of each system, which provide a basis for comparison of the two types of treatment, under identical circumstances. To the authors' knowledge, a comparison of this sort has not been reported previously.

The experience with operation of mixed bed and four bed demineralizers at the Huntley Station has led to these conclusions:

(1) Both systems can produce water of a quality suitable for makeup to high pressure boilers, but the four bed demineralizer has given more consistent operation, in this case where some organic contaminants apparently still remain, despite the pretreatment used.

(2) Although organic contaminants apparently are present, they do not seem to affect the operation of the four bed demineralizer as rapidly as the mixed bed demineralizer.

(3) Overall operating costs are less for the four bed demineralizer, but the full extent of the difference cannot be determined as yet since the resin in this system has shown no deterioration.

(4) The operators have found the four bed demineralizer somewhat more dependable in its operation, and easier to maintain.

A. K. Sukumar, Ford Motor Co., and **S. B. Applebaum,** Cochrane Corp., then gave a joint paper on "Demineralizer Treating Hot Lime Zeolite Effluent Replaces Evaporators in Ford's River Rouge Plant." This paper briefly discussed some of the past changes in the River Rouge boiler plant in the No. 1 Power House as well as in the water treatment plant that kept pace with these boilers.

The latest advance was the installation of a large automatic demineralizer (800,000 lb per hr capacity).

Blaw-Knox two-line type Clamshell Bucket design cuts handling costs

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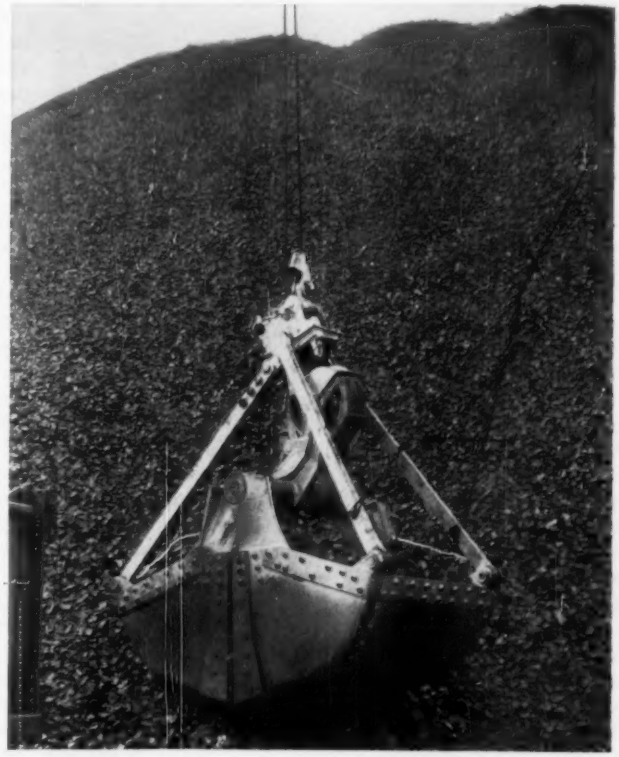

The demineralizer installation permitted the elimination of an old evaporator plant thus saving a considerable outlay of money which would have been required to rehabilitate these evaporators. At the same time it permitted the use of steam from the high pressure boilers through a spill-over valve arrangement into the low pressure plant steam system, should one of the existing low pressure boilers be forced out of service during heavy load periods.

J. A. Donald, N. A. Miller and **C. D. Blancke,** all of Sargent & Lundy, submitted a paper entitled "From Boiler Boilout to Operating Date." An individual, the authors believed, should be selected to be in overall charge of startup procedure. He should be a person of stature who will have the confidence and respect of the various groups involved with differing interests—operators, manufacturers, contractors, labor unions. His work will be largely the coordination of the activities of these groups in the most frenetic period of the construction program. This man can help greatly to reduce the startup period.

Moreover, the authors remarked, whether certain testing operations are included or excluded from the testing procedure and how they are performed can greatly affect the duration of preliminary operation. These should be carefully evaluated timewise so that an informed decision can be reached ahead of time as to what is to be done. Before the boiler boilout begins, the manufacturer and operating groups should carefully review the design, so that they will thoroughly understand all factors relating to operation.

One of the most important items, basic in obtaining continuity during initial operation, is treated water. The authors felt it must be assumed that the entire water treating plant will have been installed prior to the start of boilout, complete from initial supply through all the components of the plant; also that it will have been placed in service in its entirety and that all its individual equipment will be under automatic control.

Facilities for handling the treated water are of extreme importance and all storage facilities should be available. The water storage tanks themselves should be filled with treated water and topped off to the maximum before the start of boilout. Any miscellaneous drain tanks which are a part of the cycle should register at the full mark. The deaerating heater storage tank should be completely full and, if possible, storage should be made in the condenser to its maximum capacity. Experience has pointed out that the lack of water in certain cases has destroyed the continuity of events during startup and has delayed the entire operation. It is hard to realize how seriously and how rapidly drains or spill systems or the malfunction of equipment may deplete the water supply. Consequently, ample boiler water supply is important in attempting to establish good conditions for initial operation of the plant.

The boiler feed system is required to refill the boiler after each blowdown during boilout and, in the case of some controlled circulation boilers, to furnish injection seal water. The boiler feed pump suction line should have been thoroughly flushed and the pump placed in service prior to boilout.

Each step in preliminary operation, once begun, should be carried to completion for good results.

W. A. Crandall and **W. Nacovsky,** Consolidated Edison Co. of New York, Inc., spoke on "The Development and Operation of an Ultra Sensitive Recording Flame Photometer." The measurement of steam and water purity has always been a problem in the operation of steam power plants. Until recent years, conductivity was the accepted index of water purity. However, the continuing trend to higher and higher steam pressures has brought with it the problem of determining the composition of waters which, by conventional standards, are nearly chemically pure.

Consolidated Edison has been increasingly concerned with the performance of evaporators and high pressure boilers in respect to carryover. Data obtained by means of conductivity measurements, it has been felt, often gave an inadequate picture of the operation of the unit being tested. Certain studies have shown that in some cases such techniques as sodium flame photometry and the use of radioactive tracers produce results that can be correlated with the performance of modern high pressure boilers and evaporators.

Coal Handling

A. J. Stock, Stock Equipment Co., discussed "New Ideas for Industrial Boiler Plant Coal Handling." Today coal is definitely the least expensive fuel in many areas, and particularly in the Midwest. This situation is substantiated by data recently accumulated on the fuel, operating and maintenance costs for 38 existing boiler plants, and presented by the author in tabular form. No attempt was made to be selective except to include plants of similar size, namely up to 40 to 50 thousand pounds of steam per hour. Both old and new plants were included; coal, Mr. Stock stated, would have appeared still more favorably had the survey been restricted to plants with modern equipment. Gas fired plants were omitted since these must be equipped with additional oil or coal facilities.

The coal price shown as $7.90 per ton is actually an average of prices varying from $5.61 to $10.20 per ton, according to the plant location and the requirements of the fuel burning equipment.

It is interesting to note that if all factors of fuel, labor, supplies, auxiliary electric power, repairs and miscellaneous expenses are included, that the relative cost for Atlantic Seaboard plants burning oil would be 116 compared to 100 for coal burning plants in the Midwest. The Atlantic Seaboard is the most favorable location for oil except the oil producing areas, and the Midwest is most favorable for coal. Midwest plants burning oil would rate 140. The spread in cost can be expected to become further accentuated as the world's supply of oil is increasingly depleted.

This information, in the author's opinion, indicates that the boilers should be coal fired, except for consideration of the initial investment. The installed cost may, however, be three or four times the cost of an oil fired package boiler, if all auxiliaries, instrumentation and combustion control are included and equivalent fuel handling equipment is provided. By equivalent fuel handling for coal fired boilers is meant clean and automatic coal and ash handling Much study could profitably be devoted to the design of coal fired boiler plants,

Mr. Stock believed, and the capital costs made at least to approach those of the package oil fired unit.

By way of example the author described several coal handling arrangements for plants with truck delivery, or railroad car. In addition suggestions were made on designing silos to become a part of the building.

The utilization of an outdoor coal pile as active coal storage, as now employed by Public Service Electric and Gas Co. of New Jersey and New York Electric and Gas Corp., can also be adapted for industrial boiler plants. Such an arrangement was pictured. It suggested that a reclaim hopper, feeder, bucket elevator and long center coal scale be provided for each boiler. Interconnection among the coal elevating units provided for greater reliability. A long center scale served to transport the coal from the discharge of the bucket elevator to the boiler. These scales can be built to a length of 30 to 35 feet. If a quality scale is used, the author commented, it can be placed outdoors on the roof of the boiler auxiliary bay, thus reducing the building costs.

The arrangement as presented included a new device, called a layer loader now commercially available, for feeding coal to stoker hoppers without segregation. Basically, a layer loader is the equivalent of a swinging spout except that only the lower end of the spout is employed. This lower end becomes a car, carried on suitable wheels within a dust tight housing and moved back and fourth by a suitable drive. An automatically actuated coal valve at the inlet feeds coal to the distributing car. The layer loader gives good distribution in the stoker hopper, and it corrects segregation which may already be present in the incoming coal. The operation of the layer loader was explained by sketches.

Station Ventilation

John Dolio, Shaw, Metz & Dolio and **E. W. Landow,** Argonne National Laboratories, chose as their topic "Ventilation of a Research Nuclear Reactor Building." The design, the authors quickly stated, depends on a number of factors, the type of reactor, power of reactor, type of research work contemplated, building design and its location with respect to populated areas.

The reactor in this report is Argonne National Laboratory's CP-5, housed in Building No. 330, located on the DuPage Site near Lemont, Illinois. This building is divided into three main parts: the gas-tight section of slip form construction, a non-radioactive laboratory wing and a service and equipment storage area of conventional construction. The non-radioactive laboratory wing is ventilated through windows.

The reactor is located in the gas-tight section at the center of a cylindrical room 70 feet in diameter by 42 feet high. The entire gas-tight section of the building can be sealed by closing dampers in the supply and exhaust ducts, and by flooding water traps in the concrete section of these ducts and the escape stairways. A steel hemispherical shaped dome structure containing a flexible diaphragm has been located adjacent to the building and connected with piping to the reactor area. When the reactor area has been sealed, this structure will prevent excessive positive pressures from building up within the gas-tight section from weather changes.

R. W. Patterson, Sargent & Lundy, then described "Central Station Ventilation." A power plant, as the author stated, with its great many heat-emission surfaces and its irregular arrangement of equipment, is particularly difficult to ventilate. With a building having a relatively low heat release, large margins can be added to the air quantities to allow for unknown factors; but because of the vast quantities of air required to cool a power plant, a large margin means a large expenditure of money. This paper was offered as an attempt to establish reliable values for the factors that can be approximated so that the margin can be reduced to a minimum.

The ventilation of the major areas, in Mr. Patterson's opinion, should be designed in the following sequence: (1) choose desired maximum temperatures, (2) calculate the heat radiated by the equipment in the area, (3) calculate the volume of air required to absorb the heat and give the desired temperature rise above ambient, (4) design an air distribution pattern for the building and determine the location of air inlet and outlet openings, (5) choose the type of openings to be used and calculate the area required to pass the necessary air volume, (6) make a detailed study of particular hot spots or confined areas to determine special ventilation that might be required in the form of fans and duct work, or cooling equipment.

Editor's Note:

We regret that space limitations make it necessary for us to forego printing comments on the 3-paper session on operating experiences of Philo Station, a number of papers on fuels and fuel availability as well as other worthy individual presentations which we had prepared.

Continuous records of the oxygen dissolved in boiling feed water and of the hydrogen entrained in steam, point to the corrective measures necessary to prevent otherwise unsuspected and costly corrosion. The Cambridge Analyzers measure and record dissolved oxygen directly. The hydrogen in the steam is measure of the oxygen set free by dissociation. Cambridge Instruments are available for recording O_2 and H_2 either separately or simultaneously.

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Abstracts From the Technical Press—Abroad and Domestic

(Drawn from the Monthly Technical Bulletin, International Combustion, Ltd. London, W.C. 1)

Fuels: Sources, Properties and Preparation

Systematics of High Volatile Coals. B. K. Mazumdar. *Fuel* 1958, **37** (Jan.) 119–23.

It is shown that a direct relationship exists between the air-dried moisture, capacity moisture or heat of wetting and the calorific value of the volatiles. This can be used for the calculation of the calorific value of high volatile coals and a formula is presented taking this relationship into account.

De-sulphurization of Coke by Hydrogen. H. E. Blayden. *Coke and Gas* 1958, **20** (Jan.) 26–9, 37.

From a survey of published reports and laboratory investigations the conclusion is drawn that none of the processes proposed appears suitable for practical application.

Problems of Importing Natural Gas. A. R. Myhill. *Chem. Proc. Engng.* 1958, **39** (Jan.) 24–5.

The problems discussed are: (1) natural gas winning and transport; (2) sea transport; (3) shore reception; (4) utilisation.

Mechanical Handling

First Project for a Long Distance Coal Pipe Line. Ya. I. Balbachan. *Ugol* 1957, (Aug.) 38–40. (In Russian.)

An illustrated account is given of a project for a 61 km, 300 mm pipe line to move 200 tons/h of 0.6 mm coal from Novovolynsk mines to the Dobrotvorsk power station. The suspension of coal in water will have a solid/liquid ratio of 1 to1.6 and move at 1.46 m/sec. The coal will be fed into the pipe by a feeder having three horizontal plungers working against 70 atm pressure and at the power station will have water removed by centrifuges. The cost per ton of moving the coal 61 km is calculated to be 2.25 roubles against 5.62 for rail transport.

From *Fuel Abstracts* 1958, **23** (Jan.) 160.

Liquid Flow Transport Systems for Solids Such as Coal. Coal Industry (Patents) Ltd. and A. Whitaker. *British Patent* 788,097, 27th August, 1953.

Automatic regulating means are provided to keep the concentration of solids in the liquid at the desired ratio and also to restrict the concentration of solids when the rate of liquid flow reaches a pre-determined maximum.

Means for Use in the Hydraulic Transportation and/or Elevation of Solids. T. Moynihan, T. C. Moynihan and D. Moynihan. British Patent 789,178, 13th May, 1955.

The coal is passed into the high pressure water by the intermediary of a chamber with slide valves at top and bottom which are alternately closed and opened for feeding the coal into the chamber and from it.

Steam Generation and Power Production

Temperature Conditions of Horizontal Steam-Generating Tubes at Supercritical Pressures (230–250 atm. abs.). V. G. Chakrygin and V. A. Lokshin. *Teploenergetika* 1957, **4** (Oct.) 27–30. (in Russian.)

Results of the investigation of temperature conditions indicate that in designing once-through boiler units for extra-high steam parameters it is expedient to fix the ultimate pressure so that under all boiler operating conditions the fluid pressure in the phase transition zone is above the critical. Normal temperature conditions of the transition zone (with tubes coiled horizontally or at an angle) do not call for excessively high flow rates such as 2,000–2,500 kg/m²s. Under such conditions pearlitic steels can be used at high thermal loads (up to 300 × 10^3 kcal/m_2h) and a relatively low mean gravimetric rate (500 kg/m²s).

From *C.E.G.B. Digest* 1958, **10** (Feb. 1) 253.

Fatal Explosion of a High Pressure Boiler. H. Aull. *Techn. Überwach.* 1957, **9** (Dec.) 439–45. (In German.)

An explosion which completely destroyed a boiler and killed one man occurred in a German power station 9,000 hours after the boiler's commissioning. The boiler was rated at 250 klb/h, at 1200 psi and 975 F and equipped with tilting corner burners and a travelling-grate, the latter for operation at low loads. The steam was superheated in three stages, a horizontal superheater in the convection pass and two pendant superheaters in the pass from the furnace to the convection section. The temperature of the superheated steam was controlled by passing part of the steam from the second superheater through a cooler in the steam drum, the valves controlled the volume of steam passing to and from the drum being installed in a small, almost totally enclosed, hopper underneath the two pendant superheaters. The investigation after the accident showed that the tube connecting one of the valves with the cooler in the drum had a serious material defect, a wall thickness of only 5 mm instead of 8 mm. This tube exploded first and tore open forty-seven tubes in the rear wall of the furnace releasing so much steam into the furnace that this, too, exploded. It is stressed that all tubes for high pressure boilers should be tested by ultrasonics to detect defects, that tubes or pipelines should not be put into small closed spaces, and that systems for continuous monitoring of the temperatures at all essential parts of the boiler should form part of a boiler control scheme.

Can Hydraulic Couplings Double as Feed Water Regulators? I. J. Karassik. *Combustion* 1957, **29** (Dec.) 34–7.

Since variable speed operation of boiler feed pumps is used increasingly, the question has arisen whether it is necessary or advisable to control the flow of feed water separately where hydraulic couplings are used. The advantages and disadvantages are discussed and it is stressed that these clearly point to control by hydraulic coupling without separate regulation.

Testing Boilers for Leaks. Anon. *Engineer* 1958, **205** (Jan. 10) 71.

The "Tifa" fog-making machine introduces into the boiler after it has been completely sealed a fog generated from a material which does not sustain combustion. The pressure may be up to 0.5 in w.g. to make sure that even very fine leaks are detected. Although a boiler of 100,000 cu. ft. capacity would theoretically require 10 min. treatment, 3–4 mins. have proved adequate in the first instance with a few short bursts afterwards.

The Second Pulverised Fuel Conference. Anon. *Engng. Boil. Ho. Rev.* 1958, **73** (Jan.) 10–16.

A summary of the conference held in November 1957 is provided giving abstracts of the papers presented under the headings: (1) fuel supply background; (2) fundamental research work; (3) practical research work; (4) boiler availability; (5) developments in Great Britain; (6) low volatile coals; (7) overseas developments; (8) high ash fuels in

French power stations. A list of papers and authors is appended.

Advancements in Coal Firing Techniques. J. E. Tobey. *Coal Utilization* 1957, **11** (Dec.) 26-30.

A general review of developments during the past ten years showing that: 1. Boiler efficiency in power stations has increased from 85 to 89% with some units reaching 92%; 2. The coal rate has dropped from 1.307 to 0.938 lb/kWh and the net heat rate from 15600 to 11350 B.t.u./kWh. 3. Overall thermal efficiencies have increased from 21.8 to 31% with the most efficient units operating at 8000 B.t.u./kWh or 0.7 lb/kWh and an efficiency of 42.5%. The situation with regard to industrial boilers is less satisfactory but it is also more difficult to obtain an accurate estimate; it is believed that the average efficiency has increased by 10% in the past ten years. Advances in auxiliary equipment such as unloading and transport of coal, pulverisers and bunker design, soot blowers and combustion controls are also discussed.

Application of Oil-Firing to C.E.A. Boilers. C. Bottrell. Central Electricity Authority, undated, "Report of Conference on Oil-firing Problems, October 1956," 55–8.

Power stations involved in the C.E.A. oil-firing programme are tabulated. In addition to stations converted from coal-firing some new stations have been installed that are designed to burn oil. These stations are also designed to burn coal although initially coal handling and preparation plant will be omitted. Where possible burners in the converted boilers will be pressure atomised, and auxiliary heating systems (e.g. for oil treatment) will be independent of the main power plant. Practical steps for controlling flue gas emissions due to oil-firing have been considered by the C.E.A. and recommendations have been made to all Divisions concerned. Since two stations will be burning low flash oil special precautions in its use will have to be taken: these are outlined.

From *Fuel Abstracts* 1958, **23** (Jan.) 558.

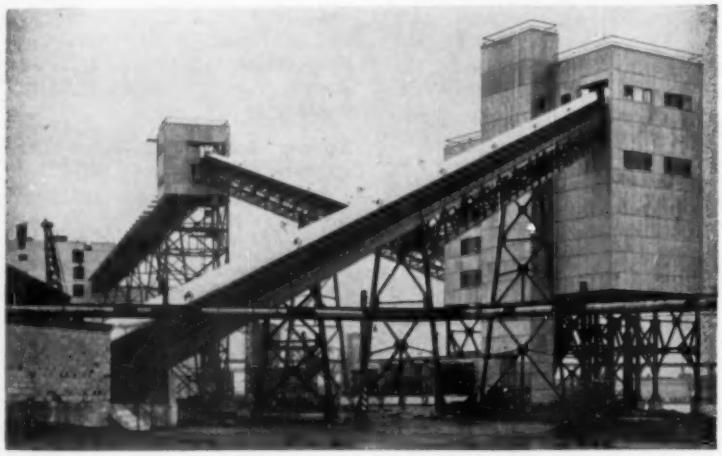
Application of Oil-Firing to C.E.A. Boilers. F. Dransfield. Central Electricity Authority, undated, "Report of Conference on Oil-firing Problems, October 1956," 59–64.

Problems associated with the conversion of pulverised fuel boilers to oil-firing are considered, and means of dealing with them are discussed. Special difficulties have been encountered in converting boilers with Bailey furnaces and corner-fired furnaces. Pressure atomisation has been used in preference to steam where possible. Other factors considered include sulphur corrosion, superheater fouling and flue dust.

From *Fuel Abstracts* 1958, **23** (Jan.) 559.

Operating Experiences at Bankside "B" Power Station. A. G. Cox. Central Electricity Authority, undated, "Report of Conference on Oil-firing Problems, October 1956," 87–97.

Fuel oil handling and heating arrangements are included in a description of boiler operation at Bankside "B" station which consists of two 60 MW turbo-alternators and four 375 klb/h oil-fired boilers. The plant is designed to handle heavy fuel oil having a viscosity up to 6500 sec., Redwood No. 1. Steam heating maintains the oil at about 110 F, and each boiler has its own pumping and heating unit. Burner equipment in each boiler consists of twelve Todd "D" type mechanical pressure atomisation burners fitted in the front wall in two horizontal rows of six. Operational experience with the four boilers is discussed; they were designed to operate with 12% carbon



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dioxide at the air heater outlet and a final gas temperature of 340 F. Apart from heavy fouling and corrosion in the primary air heater, these boilers have remained free from troublesome deposits. A flue gas washing plant controls emission of solids from the chimney.

From *Fuel Abstracts* 1958, **23** (Jan.) 560.

Operating Experiences at Marchwood Power Station. H. G. R. Jones. Central Electricity Authority, undated, "Report of Conference on Oil-firing Problems, October 1956," 105–117.

Experience with oil-fired John Thompson 550 klb/h evaporation boilers at Marchwood is described. The oil used has a viscosity of 6000 sec. with a gross calorific value of about 18,200 B.t.u./lb. Burners are arranged on either side of the boiler in four banks and are fired in pairs. Troubles causing complete loss of ignition on the boiler plant are discussed. Efforts to prevent fouling and air heater corrosion include the injection of pyridine and the proposed injection of dolomite and of ammonia. The rate of corrosion of the air heater has been measured by three different methods (scraping, insertion of weighed air heater tubes and use of a dew point meter). Stack emission has so far not been effectively controlled.

From *Fuel Abstracts* 1958, **23** (*Jan.*) 561.

Removal of Oxygen from Water and Solutions. Central Electricity Authority, E. C. Potter and G. Whitehead. British Patent 788112, 24th March, 1955.

The removal of oxygen from water or solutions is carried out in an ion exchange column, the ion exchange resin containing an insoluble metal oxide or hydroxide capable of reacting with oxygen or peroxide.

Gas-Side Corrosion and Deposits

External Fouling and Corrosion in Oil-Fired Boilers. H. E. Crossley. Central Electricity Authority, undated, "Report of Conference on Oil-firing Problems, October 1956," 143–5.

External fouling and corrosion due to sodium and sulphur in fuel oil are discussed. Sodium may cause hard acidic scale on the heating surface and the presence of vanadium may affect the acidity of these deposits. This is important in superheaters with temperatures over 1050 F. Removal of sodium from the fuel oil is difficult. Air heater corrosion is due mainly to sulphur in the fuel and the subseqent production of sulphuric acid in the boiler. Semi-reducing furnace conditions have helped to minimise sulphur trioxide production at Bankside "B" station. Other methods of reducing acid corrosion are mentioned and include the use of additives, and water washing.

From *Fuel Abstracts* 1958, **23** (Jan.) 576.

Prevention of Fouling and Corrosion. B. Lees. Central Electricity Authority, undated, "Report of Conference on Oil-firing Problems, October 1956," 146–8.

High temperature deposits have not been noticeable at Bankside "B" oil-fired power station but air heater troubles have been considerable. Attempts to prevent air heater corrosion included the injection of oil at the flame base which lowered the acid dew point temperature, and the increase of superheat surface after which the air heater showed no serious fouling during 1200 h operation. Dry cleaning of air heaters was ineffective but the cold water washing adopted every 500 h did not stop corrosion.

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Additives used with success have been tar base liquids which were injected to serve as corrosion inhibitors between primary and secondary air heaters, and soda ash, injected in the same place, to serve as a neutralizing agent. Modifications to an air heater suggest that it would be advantageous in turbular air heaters to have air inside plain tubes in a pancross pattern.

From *Fuel Abstracts* 1958, **23** (Jan.) 579.

Effect of Fuel Oil Additives on Low Temperature Corrosion and Smoke. G. F. J. Murray. Central Electricity Authority, undated, "Report of Conference on Oil-firing Problems, October 1956," 197–202.

Tests were made using different additives to determine their effect on the amount of sulphur trioxide produced and on the acid dew point. The addition of ammonia later in the system to the flue gases, where the temperature was about 350 C, gave good results. By an addition of up to 0.08% ammonia, by weight, the acid dew point was eliminated and a water dew point obtained. Corrosion using ammonia was 0.2 mg/h as compared with 6.6 mg/h without any additive. A suggested ammonia additive system is outlined. Atmospheric pollution by flue gases may also be improved due to reduction of sulphur trioxide and agglomeration of particles.

From *Fuel Abstracts* 1958, **23** (Jan.) 578.

Fuel Oil Ash Deposition and Corrosion (In Oil-Fired Boilers). T. M. B. Marshall. Central Electricity, Authority, undated, "Report of Conference on Oil-firing Problems, October 1956", 202–7.

The ash content of fuel oil may vary between 0.01 and 0.1 per cent depending on the viscosity of the fuel and its source. Aluminum, calcium, iron, magnesium, nickel, silicon, sodium, vanadium and zinc are the usual ash-forming materials present in the oil. Ash deposition on receiving surfaces below 550 C can be due to the formation of sodium sulphates as a sticky film. These deposits are not difficult to remove by soot blowing or steam lancing. Deposits formed on surfaces above 500 C are more complex and contain combined vanadium. At these temperatures sodium sulphate and vanadium pentoxide combine to form sodium vanadyl vanadates which are almost insoluble in water. Certain additives, such as zinc and magnesium may reduce the extent of ash deposition. Ash corrosion is not appreciable until temperatures of approximately 800 C are reached. Certain heat-resisting steels and high temperature alloys are affected by vanadium compounds in the molten state. Although high nickel-chromium alloys are fairly resistant to this type of attack, they are affected by sodium sulphate at similar temperatures. Cooling with oil or steam reduces ash corrosion. The use of magnesium as an additive is also beneficial since magnesium vanadate (m.p. 900 C) is formed.

From *Fuel Abstracts* 1958, **23** (Jan.) 577.

Power Generation and Power Plant

Operating Inefficiency of Industrial Boiler Plants. W. A. MacFarlane. *Pwr. and Wks. Engng.* 1958, **53** (Jan.) 4–6.

Some striking figures are given of

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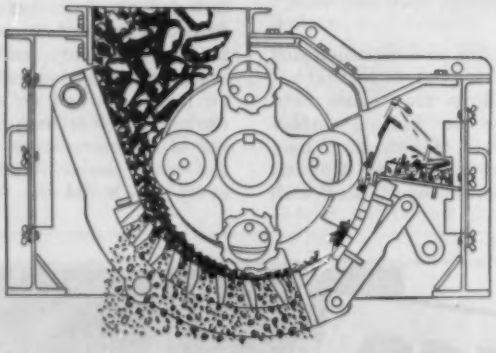

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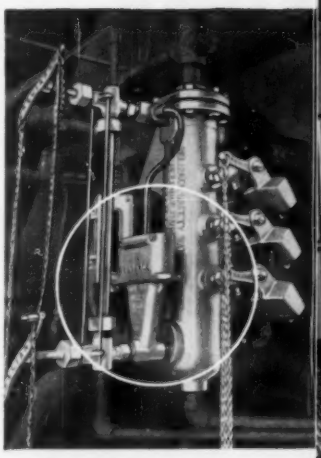
the inefficiencies at which most industrial boilers are operating as revealed by tests under standard conditions by N.I.F.E.S. The statistics over the past two years have proved that efficiencies could be improved by 15–20 per cent on average. Of 153 Lancashire boilers without economisers 90 per cent failed to reach 65 per cent efficiency, of the 402 Lancashire boilers with economisers 90 per cent failed to reach an efficiency of 75 per cent. Mechanically fired boilers had an average efficiency 3.5 per cent higher than manually fired boilers, nearly half of them exceeding 65 per cent efficiency. The CO_2 content of the flue gases were extremely low, in 10 per cent exceeding 9 per cent and in 45 per cent being less than 6 per cent; flue gas exit temperatures over 700 F were measured in boilers with economiser and over 800 F in those without. In water tube boilers, too, efficiency was very low on average, 70 per cent of the flue gases from these boilers having a CO_2 content of less than 9 per cent and 25 per cent less than 6 per cent. Economic boilers had an average efficiency of 69% but a reasonable figure would be 78 per cent.

Gas Turbine Power Stations of the Electricité de France. A. Charrier. *Techn. Moderne* 1957, **49** (Dec.) 599–618. (In French.)

The following are described: (1) the free-piston gas turbine plant at Tours containing a 6 MW gas turbine generator unit and two 7 MVar compensators; (2) the rotary compressor gas turbine plants at St. Dizier (one 6 MW generator and two 7 MVar compensators) and at Garchizy (one 6 MW generator); (3) the free-piston and compressor gas turbines, their auxiliaries and controls.

Combination of Air Turbine with Steam Process in Brown Coal Power Stations. R. Trenkler. *Mitt. V.G. B.* No. 51, 1957, (Dec.) 374–86. (In German.)

A detailed study is presented of possibilities of incorporating an air turbine in the steam generating process to improve the overall efficiency and economy of brown coal fired power stations. The assumptions on which the calculations are based are set out in full and the results presented in diagrams and tables. It is concluded that in general an increased efficiency does not result and that where relatively small increases may be expected, the higher capital costs would make the project uneconomic. This would be even more pronounced where higher grade fuels are used.



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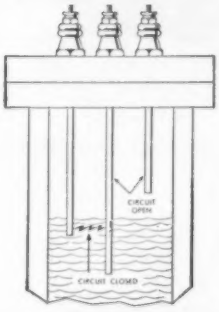

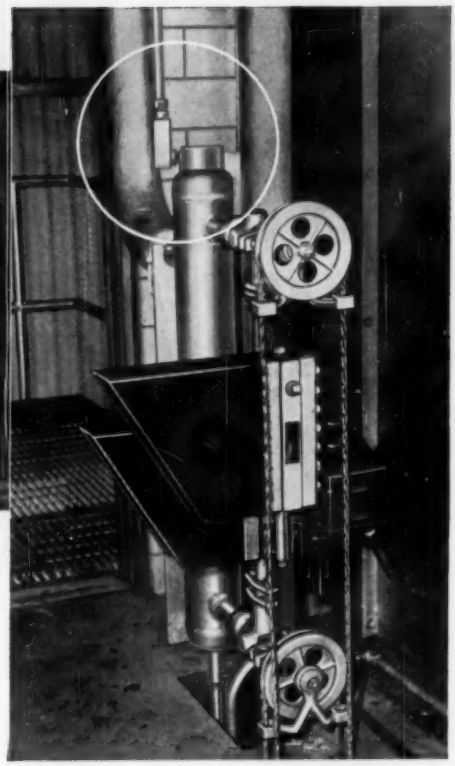

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The Time Factor in the Construction of a Thermal Power Station (with special reference to Porcheville Power Station). F. M. Lavallee and R. Minard. *Techn. Moderne* 1957, **49** (Dec.) 630–41. (In French.)

The organisation of work on the site is outlined and the conditions examined under which the main categories of work were carried out, such as foundations, building frames, boiler installation and control room. The procedures adopted for cleaning water and steam tubes and the boiler as a whole are described in detail. The saving in time possible by the correct planning of all operations and their proper sequence is discussed. In the case of Porcheville station it has taken twenty-three months from cutting the first sod to connecting the generator to the grid.

Once-Thru Boiler Used for Subcritical Operation. H. E. Gismond. *Elect. World* 1957, **148** (Dec. 23) 52–4.

The Frank M. Tait station is to be modernised by the removal of several old boilers and turbines and the installation of two 130 MW units. The boilers for these are of the C.E. pulverised coal fired once-through type rated at 940 klb/h at 2400 psi and 1050/1000 F so that both the makers and the user can obtain experience before proceeding to supercritical pressure. Feed water conditions laid down are: (1) total solids content 0.5 ppm max.; (2) pH 8.8–9.2; (3) slight excess of hydrazine; (4) dissolved oxygen 0.007 ppm. The control system for feed water flow, primary steam temperature and the turbine by-pass system will be provided by Sulzer, the combustion controls by Leeds & Northrup.

Shippingport Puts Out Full Power. Anon. *Elect. World* 1957, **148** (Dec. 30) 52.

This reactor was started up on December 18th, 1957, and by December 23rd it had reached its full output of 68 MW gross, 8 MW of which is required for auxiliary power. The total cost is estimated at $72.5 mil., of which $55 mil. was required for the reactor itself, the remainder for the conventional part of the plant. The reactor is cooled by light water at a pressure of 2000 psi, an inlet temperature of 508 F and an exit temperature of 542 F and has four cooling loops, three of which are needed to produce 60 MW net.

Nuclear Energy

Controlled Thermonuclear Processes. Anon. *Engineer* 1958, **205** (Jan. 31) 160–4.

Detailed descriptions are given of "Zeta" (Zero Energy Thermal Apparatus) and "Sceptre III" in which temperatures of 5×10^6 and 4×10^6 C respectively have been achieved in fusion reactions. Sir John Cockroft's statement of the earlier and probable future developments of these apparatus and the likelihood of the application of controlled fusion reactions to power generation is reproduced.

Future Engineering Development of the Gas-Cooled Nuclear Reactor. J. M. Kay. *J. Iron and Steel Inst.* 1958, **188** (Jan.) 3–8.

The basic technology of the gas-cooled graphite moderated reactor is reviewed and the means by which further progress may be achieved.

How Tubes Compare for Nuclear Reactor Boilers. P. S. Otten. *Power* 1958, **102** (Jan.) 81–3, 176, 178, 180.

The effect of various reactor gaseous or liquid cooling media on tube materials is considered. In gas-cooled reactor systems carbon steel can be used for temperatures below 800 F, $2^1/_4$ per cent Cr–1 per cent Mo steels for up to 1100 F and stainless steel and Inconel for temperatures between 1000 and 1500 F.

Analysis and Testing, Research

Pinpointing Boiler-Tube Corrosion. R. N. Peterson. *Power* 1958, **102** (Jan.) 112–3.

A brief account is given of the techniques evolved for inspection of the tubes of a cyclone furnace by radioisotopes; with modifications the techniques can also be applied to water wall sections of boilers. In the cyclone furnace the probe was inserted along the centre line of the furnace and the film wrapped around the outside. The photographs clearly showed which tubes to replaced.

Economics and Statistics

A Decade of Change. W. Bradbury. *Coal Utilization* 1957, **11** (Dec.) 22–4.

The development of coal consumption in U.S.A. between 1947 and 1957 is reviewed. 1947 was the peak year of consumption with 615 mil. tons, from which it gradually declined to 394 mil. tons in 1954, largely because of the dieselization of the railways and increased use of oil and gas by industrial and domestic consumers. Consumption has risen again since then to 500 mil. tons because of increased demand by power stations and export, whilst railway and industrial consumption declined further and that of the steel industry remained static.

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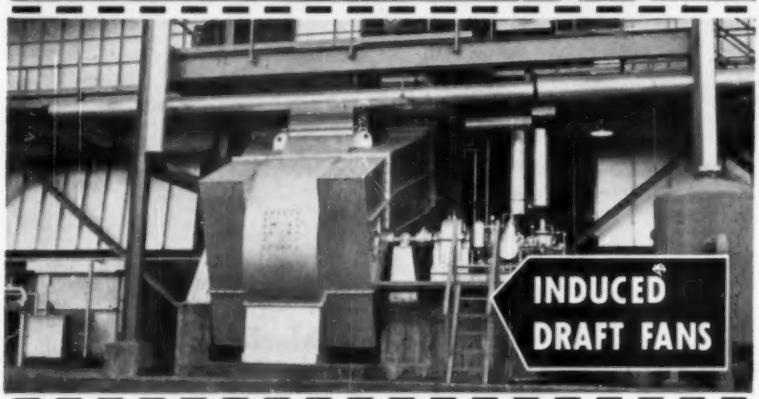

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Advertisers' Index

Aerotec Corporation, The..... *
Air Preheater Corporation, The..... 66
Alco Products, Inc..... *
American-Standard, American Blower Division..... 4

Bailey Meter Company..... 34
Ballard, Sprague & Thomas Inc..... 69
Baltimore & Ohio Railroad.... *
Bayer Company, The..... 14
Bituminous Coal Institute .....22 and 23
Bird-Archer Company..... 75
Blaw-Knox Co., Blaw-Knox Equipment Div..... 63
Buell Engineering Company, Inc..... 36

Cambridge Instrument Company..... 65
Carborundum Company, The. *
Chesapeake & Ohio Railway... *
Clarage Fan Company..... 76
Cochrane Corporation..... *
Combustion Engineering, Inc. .....Second Cover, 10 and 11
Combustion Publishing Company, Inc..... *
Copes-Vulcan Div., Blaw-Knox Company.....6 and 7
Crane Company..... 17
Croll-Reynolds Engineering Co., Inc..... 30

Dampney Company, The..... 27
Dearborn Chemical Company. 21
De Laval Steam Turbine Company..... 13
Diamond Power Specialty Corporation.....Third Cover
Dowell Inc..... *

Eastern Gas & Fuel Associates. *
Economy Pumps, Inc..... 2
Edwards Valves, Inc.....58 and 59
Engineer Co., The..... 69

Fairmount Chemical Co., Inc. 68
Fly Ash Arrestor Corporation.. 74

Graver Water Conditioning Company..... *

Hagan Chemicals & Controls, Inc..... 52
Harbison-Walker Refractories Company..... *

(Continued on page 75)

Ingersoll-Rand Company..... *

M. W. Kellogg Company, The.. 5
Koppers Company, The..... 16

Leeds & Northrup Company... *

Manning, Maxwell & Moore, Inc..... 26
Maryland Shipbuilding & Drydock Company..... 35
W. K. Mitchell & Company.... *

National Aluminate Corporation..... *
Norfolk and Western Railway.. 20

Pacific Pumps, Inc..... *
Peabody Engineering Corporation..... *
Pennsylvania Crusher Div., Bath Iron Works Corp..... 70
Pittsburgh Piping & Equipment Company..... *
Powell Valves..... 24
Prat-Daniel Corporation..... *
Henry Pratt Company..... 12

Refractory & Insulation Corp.. 72
Reliance Gauge Column Company, The..... 71
Republic Flow Meters Company.....32 and 33
Republic Steel Corporation ..... 28 and 29
Research-Cottrell, Inc. 18 and 19
Richardson Scale Company... 73
Rohm & Haas Company..... *

Stock Equipment Company... 31
Sy-Co Corporation..... 68

W. A. Taylor and Company.... *
Todd Shipyards Corp., Products Div..... *

Unafrax Construction Company..... *

Walworth Company.....8 and 9
Western Precipitation Corporation.....Fourth Cover
Westinghouse Electric Corp.... 25
Westinghouse Electric Corp., Sturtevant Div..... *
C. H. Wheeler Mfg. Company.. 2
Worthington Corporation..... 15

Yarnall-Waring Company..... 3
Yuba Consolidated Industries, Yuba Heat Transfer Div..... 60

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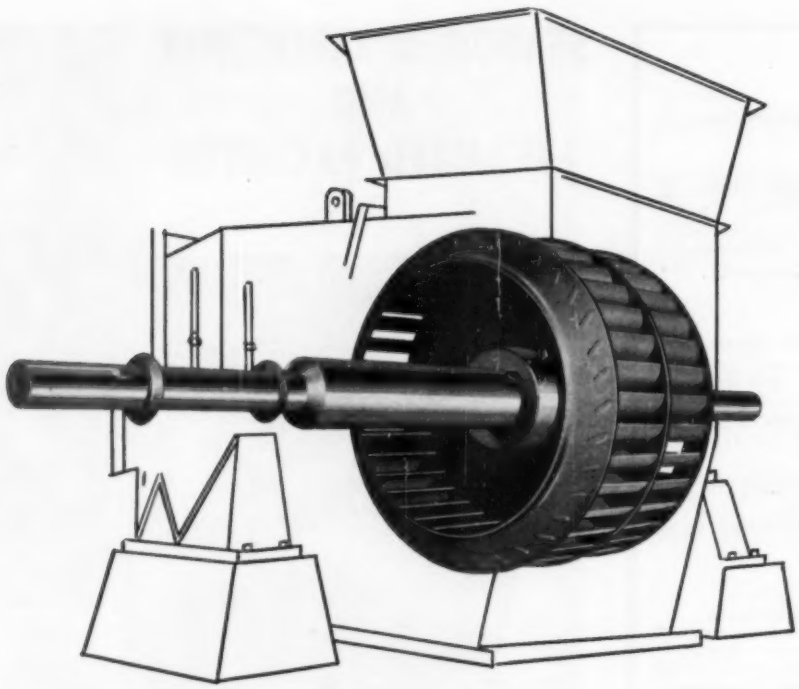

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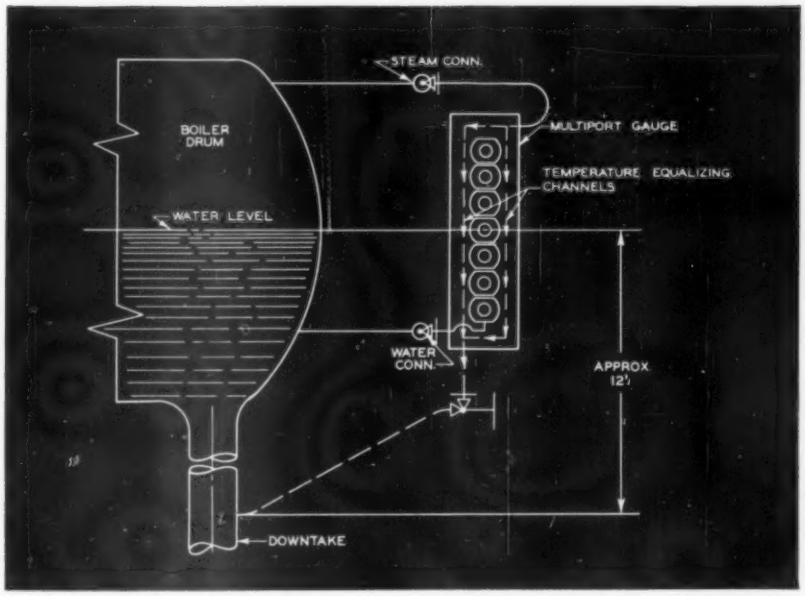

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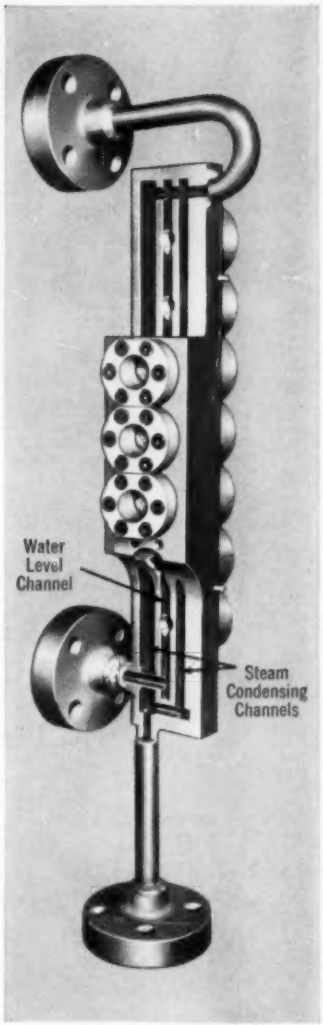

Cut-away sections illustrate independent gauge body heating channels of the new Diamond Temperature-Equalizing Multi-Port Gauge.

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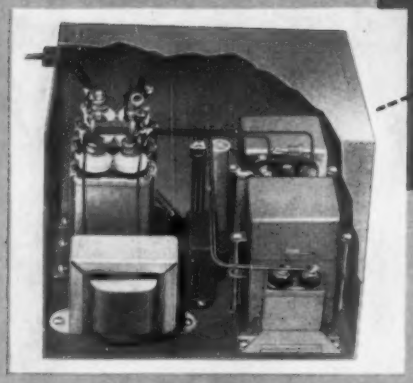

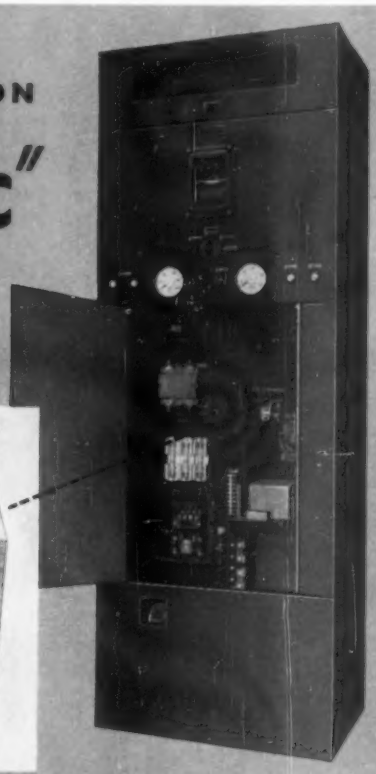

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